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AD365464	
CLASSIFICATION CHANGES	
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USAAMRDL ltr 13 May 1964; USAAMRDL ltr 13 May 1964	

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USAAVLABS TECHNICAL REPORT 65-49

(C) POWERPLANT STUDIES FOR SHAFT-DRIVEN HELICOPTER (U)

FINAL REPORT

EDR 4110

July 1965

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-213(T)
ALLISON DIVISION - GENERAL MOTORS

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DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS. VIRGINIA 23604

(U) This report has been reviewed by the U. S. Army Aviation Materiel Laboratories and is considered to be technically feasible. The results are published for the exchange of information and stimulation of ideas.

(U) For the specific missions given, the results show what effects various engine configurations and multiengine clustering concepts and related drive systems have on the gross weight, the fuel consumed, and the range of a heavy lift helicopter.

(U) Representative values were used by the contractor in establishing vehicle base line weights and propulsion system installed performance. USAAVLABS supplied estimated power-required curves for use in calculating vehicle performance.

(U) Particular attention should be given to the effects of regeneration, since it was concluded that the regenerative cycle engine would save fuel for all three missions.

(U) The specific weights of the main power transmissions, as estimated by the contractor and as used in this study, appear to be slightly optimistic for the vehicle projected time frame.

Task 1M121401D14415
Contract DA 44-177-AMC-213(T)
USAAVLABS Technical Report 65-49
July 1965

(C) POWERPLANT STUDIES FOR SHAFT-DRIVEN HELICOPTER (U)

Final Report

EDR 4110

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Prepared by

Allison Division - General Motors
Indianapolis, Indiana

for

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

(U) ABSTRACT

Powerplant studies were conducted for a shaft-driven, heavy-lift heli-
copter (HLH) with design gross weights ranging from 75,000 to 85,000
pounds and a maximum payload capability of 40,000 pounds. The num-
ber of engines required, the gross weight, and the fuel consumed for
specified missions using various powerplant configurations were deter-
mined. Control system studies were conducted to investigate control
modes, power train dynamics, and multiengine control and load shar-
ing. Power coupling studies were conducted which included combining
and reduction gearing, cross shafting, clutches, and gas-coupled rotor
drives.

The results of these studies indicate: use of regenerative engines will
result in fuel savings on all missions; approximately 18,000 shaft
horsepower (sea level static rating) may be required; current multi-
engine control techniques are applicable to the HLH; and a gas-coupled
rotor drive system warrants further investigation.

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(U) PREFACE

This is the final report on the Allison study project entitled "Power-plant Studies for a Shaft-Driven Helicopter." This study was conducted for the U. S. Army Aviation Materiel Laboratories (USAAML)* under contract DA 44-177-AMC-213 (T). These studies were conducted during the 6-month period from 30 June through 31 December 1964.

USAAML technical direction was provided by C. H. Carper, Jr., with D. S. Monson serving as the Allison project engineer. The principal investigators at Allison were R. L. Eckman, C. M. Hawkins, R. M. Swick, and J. R. Wooten.

The data and information obtained from the Sikorsky Aircraft Division, United Aircraft Corporation, Vertol Division, The Boeing Company, and the Kaman Aircraft Corporation are gratefully acknowledged.

Certain sections of this report have been classified Confidential (Group 3) because T78 engine performance data are included. These sections have been appropriately identified.

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*Abbreviation for U. S. Army Materiel Laboratories has been changed to USAAVLABS.

(U) TABLE OF CONTENTS

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(U) LIST OF TABLES

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(C) SUMMARY (U)

(U) This report presents the results of powerplant studies for a shaft-driven heavy-lift helicopter (HLH). Various powerplant concepts and configurations were evaluated based primarily on weight savings and performance. In addition, powerplant control and power transmission problems were investigated. The results of the study are summarized in the following paragraphs.

(C) Use of regenerative cycle engines will result in a reduction of lift-off weight or extension of range for the ferry mission. With regenerative engines, the 1500-nautical-mile ferry mission can be accomplished with an initial gross weight 18 percent below that required for the best nonregenerative free turbine configuration considered. A regenerative-engine-powered HLH with a ferry mission design load factor of 2.0 would have a ferry range of approximately 2012 nautical miles, as compared to approximately 1514 nautical miles for a nonregenerative powered vehicle.

(C) Use of regeneration will result in fuel savings on all missions considered. Fuel savings of 21,000 pounds for the ferry mission, 1760 pounds for the transport mission, and 570 pounds for the heavy-lift mission can be realized. This fuel savings can be an important factor when system logistics are considered.

(U) The results of the regenerator optimization study indicate that the regenerator now being developed for the Model T78 represents a reasonable compromise if all missions have equal importance. From a gross weight viewpoint, regeneration offers little to no advantage for the transport or heavy-lift missions studied.

(U) The results of the mission analysis indicate that extended range or lower fuel consumption can be attained by selective shutdown of engines during the transport and ferry missions. This can be done safely and will result in ferry mission lift-off weight and fuel savings of up to 13,000 pounds.

(C) The size of the HLH powerplant and, consequently, the number of engines required for a given output, is dictated by the 6000-foot, 95°F. transport hover requirement. Based on the power requirement curves established for this study and a heavy-lift mission empty weight to design gross weight ratio of 0.47, the HLH powerplant must be rated at approximately 18,000 shaft horsepower (sea level static ARDC standard

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ambient conditions). Therefore, four engines rated at 4500 shaft horsepower or three engines rated at 6000 shaft horsepower would be required unless some means of power augmentation is employed. On this basis, four Allison Model 546, 548, and 501-M26 engines, or three Allison Model 501-M25 engines would be required.

(U) The possibility of "flat rating" or limiting the power output of the engines as a function of ambient pressure and temperature should be considered. This would limit the maximum power input to the transmission, which would result in a lighter weight design while providing improved hot-day and altitude performance.

(U) The gas-coupled rotor drive system shows promise of eliminating the problems associated with right angle drive (bevel gear) power transmission while increasing installation flexibility. This concept provides for vertical installation of the power turbines on the transmission, thereby eliminating bevel gearing. Performance and preliminary design studies indicate that the multiple turbine approach is superior to the single-turbine approach.

(U) The weight of the gas-coupled rotor drive system preliminary design resulting from this study is estimated to be approximately 6600 pounds. This weight includes four gas-producers, ducting, four power turbines, and the transmission. The ratio of the powerplant weight to vehicle design gross weight compares very favorably with that of much lighter vehicles.

(U) The clutch design studies and operating experience indicate that a lightweight, hydraulically operated clutch capable of transmitting 5000 horsepower is feasible and can be successfully developed. The unique design resulting from this study is sensitive to reverse torque and disengages automatically in case of engine malfunction or programmed underspeed, thereby eliminating the need for an overrunning clutch. Except for the operation of one switch which commands engage or disengage, all sequencing in the clutch assembly, including the control of cooling oil, is accomplished automatically.

(U) Multiengine power-combining and speed-reduction functions should be integrated into the transmission. The elimination of casings, gears, bearings, and seals results in a simpler and lighter weight system.

(U) The use of supercritical shafting offers a means of reducing shafting weight, simplifying the installation, and improving reliability.

(U) The fuel-control systems of all engines studied are readily adaptable to the multiengine HLH powerplant.

(U) Adequate control of the load distribution between the various engines of the powerplant can be accomplished by using an open-loop system. Based on variations in power output and controls of current production engines, a total variation of 5.0 and 8.0 percent between engines could be expected at maximum and minimum power, provided that a single governor system is used. If closer load sharing control is desired, a torque-sensing, closed-loop system could be provided at the expense of added complexity. From an engine and transmission protection viewpoint, and open-loop system would be completely satisfactory.

(U) Pilot control of the multiengine powerplant can be readily accomplished using a single "twist grip" on the collective pitch lever. The control system would provide automatic rotor and powerplant coordination. A mechanical link would be provided between the "twist grip" and the individual controls for manual power modulation in the event of automatic control malfunction.

(U) From an operational viewpoint, the free turbine engine appears to be better suited for helicopter use than the fixed turbine engine. This results from the torque-speed relationship and the maneuvers required of current helicopters. The operational modes, the increased size, and the redundant engine features of the HLH could affect this conclusion.

(U) Analog studies indicate that power train torsional stability should not be a significant problem with the HLH.

(U) Performance and installation studies also indicate that either the Model T56 or T78 engine types are uniquely suited for the HLH powerplant.

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(U) RECOMMENDATIONS

GAS-COUPLED ROTOR DRIVE

Preliminary design studies have indicated that the Allison gas-coupled rotor drive system offers a solution to the mechanical right-angle drive problem while providing installation flexibility and a potential reduction in the combined power generation and transmission system weight. Detailed transmission design and powerplant-vehicle integration studies to explore the full potential of this approach are recommended.

LOGISTICS AND COST EFFECTIVENESS

Substantial fuel savings can be realized using regenerative engines; therefore, studies should be conducted to determine the effect of this fuel savings on total annual fuel cost and other systems cost. Such studies should include projected force levels under various postulated training and combat conditions, including deployments to remote areas.

SUPERCRITICAL SHAFTING

The use of supercritical shafting has the potential for simplifying and increasing the reliability of interconnective shafting systems by greatly reducing the number of bearings and spline joints and reducing the loading on the remaining bearings. In addition, a significant weight savings can be achieved with such a system. It is recommended that full-scale testing of shafting sized for the HLH be initiated. Such tests should include simulation of loads imposed by airframe dynamics.

MISSION IMPORTANCE

It is recommended that the relative importance of each anticipated HLH mission be established so that the optimum powerplant can be defined.

REGENERATIVE ENGINES

Regenerative engines should be considered for use in other helicopters as well as the HLH. The flat specific fuel consumption versus shaft horsepower curve of this type engine permits installation of higher powered engines without a sacrifice in cruise economy. This could result in improved altitude and hot day performance and an extension in helicopter range.

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(U) INTRODUCTION

This study resulted from an Army aviation requirement to investigate various powerplant concepts and multiengine configurations for a shaft-driven HLH in the 75,000- to 85,000-pound range. The objective of the study was to develop sufficient data to permit a preliminary evaluation of several HLH concepts and configurations. The study was limited to growth versions of existing engines or engines currently under development. The evaluation was based primarily on weight savings (engine installation plus fuel) and performance.

Four basic missions were defined by the U. S. Army Aviation Materiel Laboratories (USAAML)*-transport, heavy lift, ferry, and special heavy lift.

These missions, described in Table 1, were to be considered of equal importance.

For the purpose of this study, USAAML furnished power requirement estimates as a function of gross weight, velocity, and equivalent flat plate frontal area. These data are presented in Figure 1. The study consisted of performance, control system, and power coupling analyses.

TABLE 1
HLH MISSIONS

	Transport	Heavy Lift	Ferry	Special Heavy Lift
Payload (Outbound), tons	12	20	0	16
Distance, nautical miles	100 (radius)	20 (radius)	1500 (range)	100 (radius)
Cruise Velocity (With Payload), knots	110	95	Best for Range	95
Cruise Velocity (Without Payload), knots	130	130		130
Cruise Altitude	SLS	SLS	SLS	SLS
Hover Capability (OGE)	6000 feet, 95°F.	SLS	—	SLS
Hovering Time, minutes	3 at takeoff	5 at takeoff	—	5 at takeoff
Notes:	2 at midpoint (with payload)	10 at destination (with payload)		10 at destination (with payload)

Reserve Fuel—10 percent of initial fuel for all missions
Fuel Allowance—for start, warm-up, and takeoff per MIL-C-5011A for all missions
SLS—Sea Level Standard
OGE—Out of Ground Effect

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*In June 1965, the abbreviation was changed to USAAVLABS.

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The performance analysis consisted of establishing the performance for various fixed-shaft and free-turbine engines, the number of engines required, and the effect of various engine configuration and operational methods on the gross weight and fuel consumption for each mission. In addition, optimization studies were conducted on regenerators, remote gas-coupled power turbines were analyzed, and methods of hot-day power augmentation were investigated.
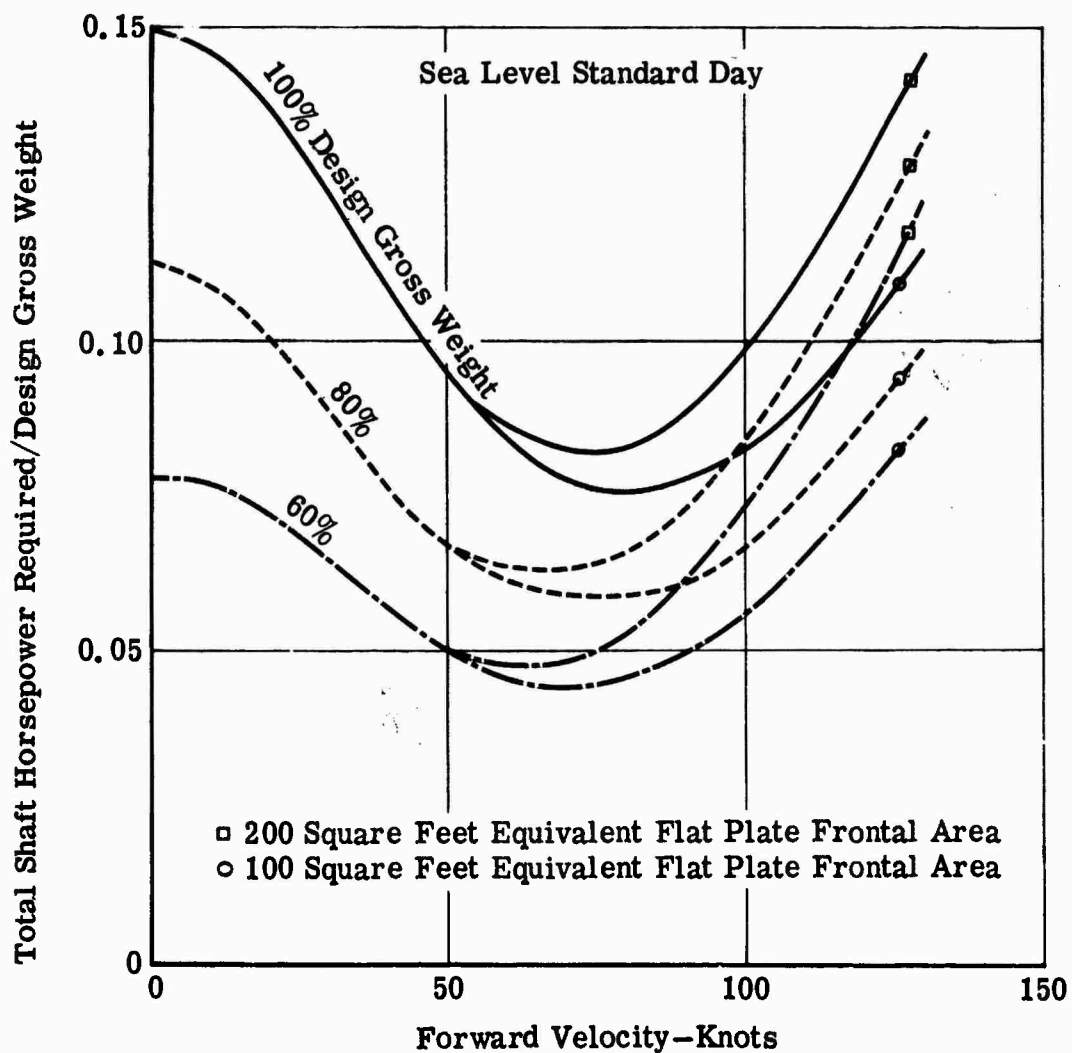
Figure 1. Shaft Horsepower Required for HLH

The control system analysis included an assessment of free-turbine versus fixed-turbine engines, transient and stability stuales, control mode analysis, starting investigation, and load sharing analysis. Engine-out problems, engine-rotor coordination, and hot-day power augmentation were also considered.

The power coupling analysis was concerned with combining and transmitting the power produced by the multiengine installations. These studies were conducted on combining gearing, reduction gearing, clutches, shafting, and safety devices. In addition, a study was conducted on a gas-coupled, integrated turbine-transmission rotor drive.

(C) PERFORMANCE STUDIES (U)

(U) Performance studies were concerned with two broad categories—engine performance and mission performance. Engine performance studies were concerned with the power output and fuel consumption for the various configurations at sea level standard conditions and at 6000 foot, 95°F. ambient conditions. The effect of velocity on performance over a range of 0 to 300 knots was also determined. In addition, methods of increasing or augmenting the power output for hot-day operation were considered. Regenerator designs and characteristics were investigated in an effort to determine optimum regenerator size for each mission.

(U) Data developed from engine performance studies were applied in the mission studies to determine the number of engines required, fuel load, and takeoff gross weight for the various HLH missions. The effect of deliberate engine shutdown on mission fuel consumption was also determined.

(U) FIXED- AND FREE-TURBINE ENGINES CONSIDERED

(U) Several derivations of the Allison T78 and T56 engines were considered in this study. These engines are briefly described in the following paragraphs—additional descriptive information is presented in the section titled Installation Studies.

(U) T78 Derivative Models

(U) The basic Model T78 engine has an axial-flow-type gas generator consisting of a 14-stage compressor and a 4-stage turbine. The compressor incorporates variable stator geometry in the inlet guide vanes and the first six stages of compression and has a design pressure ratio of 11.7. The turbine unit is designed to permit cruising at elevated turbine inlet temperatures for high thermal efficiency and low specific fuel consumption—this is made possible by air-cooling the first- and second-stage turbine blades and vanes. A tube-and-shell-type regenerator is used to reduce exhaust heat loss, thereby providing reduced fuel consumption, especially at reduced power settings. Maximum use is made of lightweight materials and high-efficiency components.

(U) The following engines were derived from the basic Model T78 regenerative turboprop engine configuration:

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- 545-C2—This engine is a front-drive regenerative turboshaft with a variable speed operating mode and constant turbine inlet temperature. The design speed for this engine is 19,300 r. p. m. with an operating speed modulation capability down to 75 percent r. p. m. No reduction gear is provided.
- 545-C3—This engine is similar to the Model 545-C2, with the exception of the engine control which is modified to provide a constant speed (19,300 r. p. m.) and turbine inlet temperature mode of operation. Power is varied by changing the compressor variable vane settings.
- 546-C2—This engine is a nonregenerative turboshaft version of the Model T78 engine. It has a variable speed and variable turbine inlet temperature operating mode. No reduction gear is provided.
- 546-C3—This engine is identical to the Model 546-C2 except a constant speed, variable turbine inlet temperature operating mode is used.
- 548-C2—This engine is a nonregenerative turboshaft derivative of the Model T78 engine. It has a rear-drive, free power turbine and can be provided with either direct drive or an integral 3.22 reduction gear.
- 548-RT—This engine is a modification of the Model 548-C2 with the free turbine removed and relocated on the main transmission.

(U) T56 Derivative Models

(U) The following engines were derived from the basic Model T56 turboprop engine configuration:

- 501-M25—This is a front-drive turboshaft engine growth version of the Model T56 engine. The compressor incorporates variable geometry (inlet vanes plus the first five stator vane stages). A short, annular combustion chamber and an internal fuel manifold are used. The turbine has four stages—the first two are air-cooled (blades and vanes). A single or bifurcated large area exhaust nozzle can be provided. This configuration also includes a self-contained oil system and a Model T56 torquemeter.
- 501-M26—This is a rear-drive, free turboshaft engine. The compressor incorporates variable geometry (inlet vanes plus the first five stator vane stages). A short, annular combustion chamber and an internal fuel manifold are used. The power turbine is gas-coupled to the two-stage gas producer turbine. This engine also has a self-contained oil system.

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(U) JET NOZZLE OPTIMIZATION

(U) Prior to developing performance data, it was necessary to determine the jet nozzle area for the engines being considered for the HLH application. These data were required since the jet nozzle area of the baseline engines had been sized for high airspeed performance. With most helicopter missions at low velocity, performance can be improved by increasing the jet nozzle area. It was found that by making the nozzle diameter for the 545 engines approximately the same as the outside diameter of the turbine and the inside diameter of the regenerator (300-square-inch area), good gains in performance could be obtained by both the regenerative and nonregenerative engines at military and normal power. Figures 2 and 3 show the effect of jet nozzle area. A further increase in nozzle area produces smaller gains. Since a considerable portion of the missions will be at reduced power, the jet velocity with the larger areas may become less than the forward velocity, producing additional drag. Therefore, a jet nozzle area of 300 square inches was used in the calculation of performance for the Model 545, 546, and 548 engines.
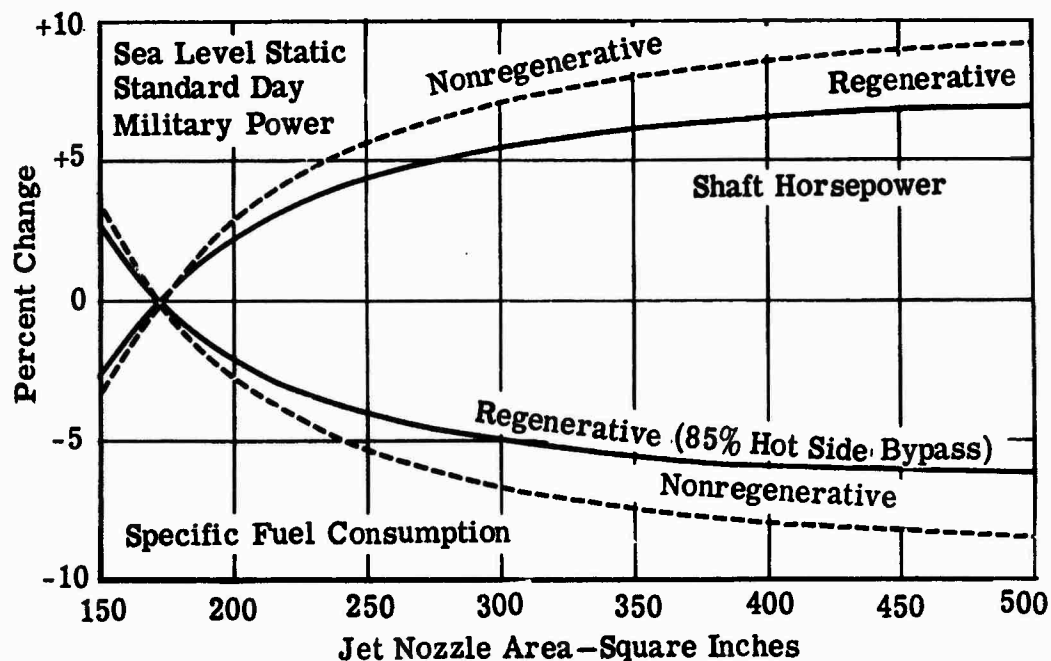
Figure 2. (U) Effect of Jet Nozzle Area on T78 Performance

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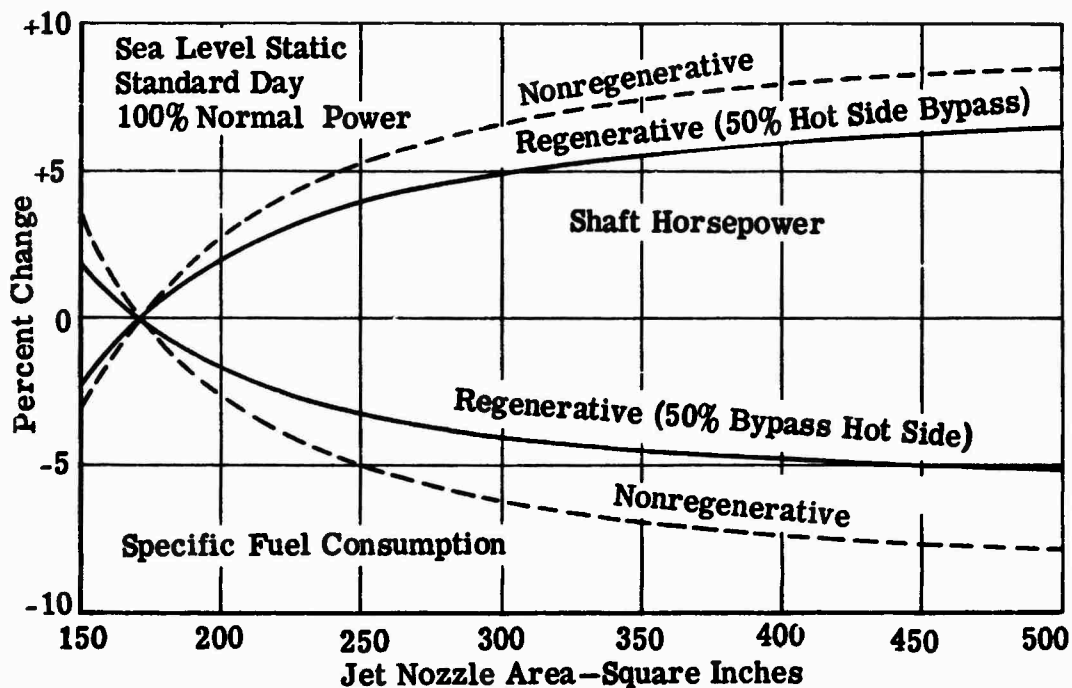
Figure 3. (U) Effect of Jet Nozzle Area on T78 Performance

(U) A similar study was conducted for the T56 derivative Models 501-M25 and 501-M26. As shown in Figure 4, the slope of the adjustment curves becomes relatively flat above 500 square inches. Therefore, 500 square inches was selected as the jet nozzle area for the Model 501-M25 and 501-M26 engines.

(C) PERFORMANCE ANALYSIS (U)

(U) Performance data were developed for a number of engine models. These data include the effect of increased jet nozzle area as previously discussed. All data are based on sea level ARDC standard ambient conditions unless otherwise noted. Table 2 is included to aid the reader in obtaining a summary of the pertinent performance data for the engines considered.

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Sea Level Static
Standard Day
Military Power
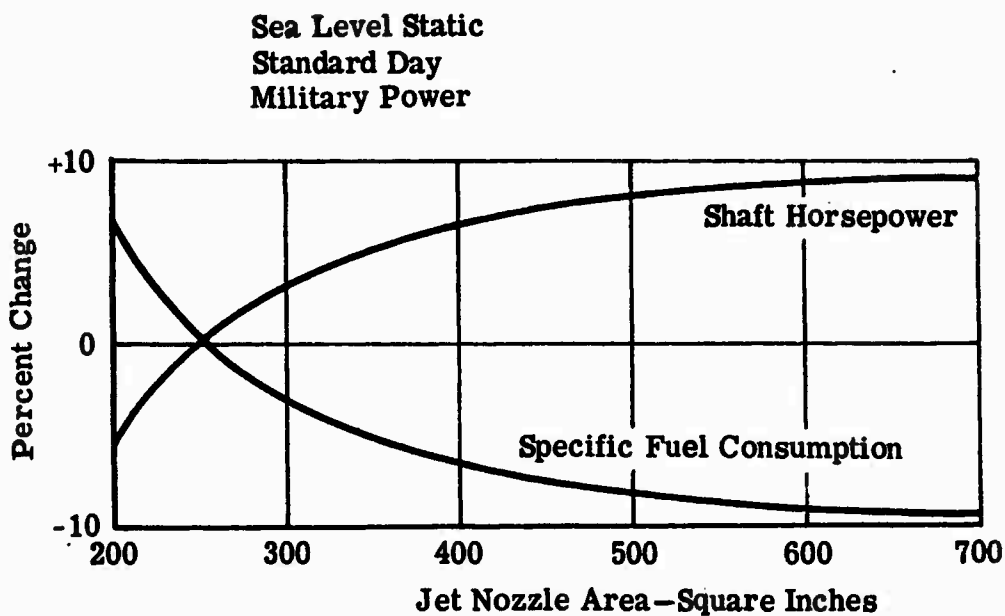
Figure 4. (U) Effect of Jet Nozzle Area on 501-M25
and -M26 Performance

TABLE 2
(C) ENGINE PERFORMANCE SUMMARY (U)

Model Designation	Turbine Type	Regenerative	Weight (lb)	Military Performance At Sea Level Static		Performance At 6000 Feet, 95°F.	
				SHP	SFC	SHP	SFC
545-C2	Fixed	Yes	1141	4105	0.503	2865	0.518
545-C3	Fixed	Yes	1141	4105	0.503	2865	0.518
546-C3	Fixed	No	681	4511	0.479	3128	0.496
548-C2	Free	No	822	4489	0.482	3080	0.501
548-RT	Free	No	—	4354	0.496	2979	0.518
501-M25	Fixed	No	955	6000	0.500	4075	0.521
501-M26	Free	No	1030	5450	0.479	3715	0.500

Note:

SHP—shaft horsepower
SFC—specific fuel consumption

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(U) Performance was determined for the 545 (regenerative) engine series for various operating modes. As shown in Figure 5, the performance attainable at constant r. p. m. and turbine inlet temperature (T_4) with variable airflow is superior to that with constant r. p. m. and variable T_4. For this reason, the latter operational mode was not considered for further study. The best performance was shown with the T78 operational mode—variable r. p. m. and constant T_4 (545-C2). Use of this mode would require rotor speed variation from 100 percent at military power down to 75 percent at minimum power.
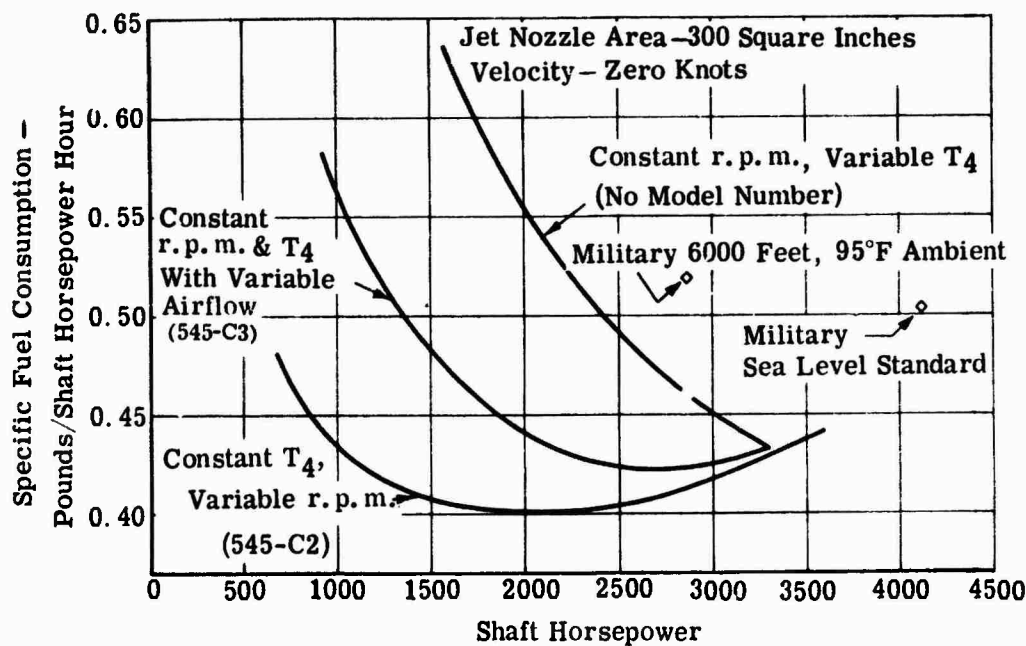
Figure 5. (C) Model 545 Regenerative Turboshaft
Engine Performance (U)

(U) The data shown on Figure 5 are for static conditions. To conduct the mission analysis, the effect of velocity on performance must be considered. Figures 6 and 7 present fuel flow and shaft horsepower for various true airspeeds for the Model 545-C2 and 545-C3 engines. These and all curves presented in this section of the report do not include performance penalties due to installation effects or specific fuel consumption depreciation, since these factors differ for various installations and applications. (These penalties are taken into consideration in the subsection titled Mission Analysis.)

(U) The performance of the Model 546 engine with various operational modes is shown in Figure 8. A comparison of Figures 6 and 8 shows that the operational mode has much less effect on nonregenerative engine performance. The effect of operational mode only begins to become significant at power outputs much lower than those anticipated for helicopter operation. The effect of velocity on fuel consumption and shaft horsepower for the Model 546 series engines is shown in Figures 9 and 10.

(U) Performance data for the Model 548-C2 nonregenerative, free-turbine engine are shown in Figure 11. The effect of velocity on fuel consumption and shaft horsepower is shown in Figure 12.

(U) The gas producer section of the Model 548-C2 engine is suitable for driving remote turbines and supplying gas for hot-cycle systems. The gas flow, fuel flow, and gas producer outlet temperature are presented as a function of discharge pressure in Figure 13.

(U) Direct vertical installation of single and multiple power turbines on the main power transmission was investigated. (The mechanical aspects of this study are reported in the section titled Power Coupling Studies.) From a performance viewpoint, there are some serious reservations regarding the multiple gas producer-single turbine approach. An annular common inlet plenum can be quickly eliminated by considering the effect of engine loss on power output. It was found that if one gas producer fails or is shut down in a three-engine installation, a 58-percent loss in shaft output power would result. Loss of one gas producer in a four-engine installation would result in a power loss of 48 percent. The power loss is the result of loss of flow, compounded by the loss in expansion ratio.
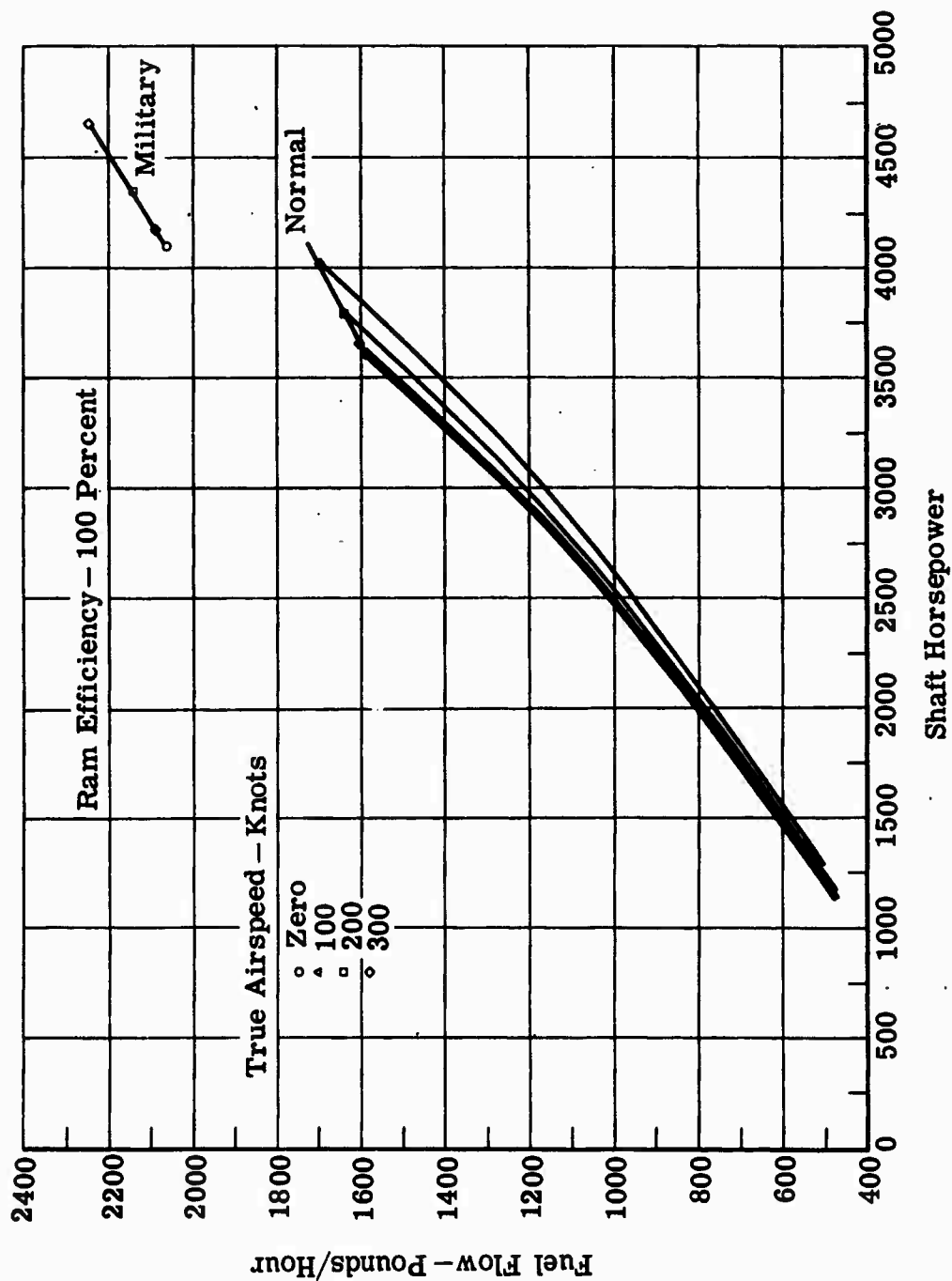
Figure 6. (C) Estimated Performance—Model 545-C2 (U)

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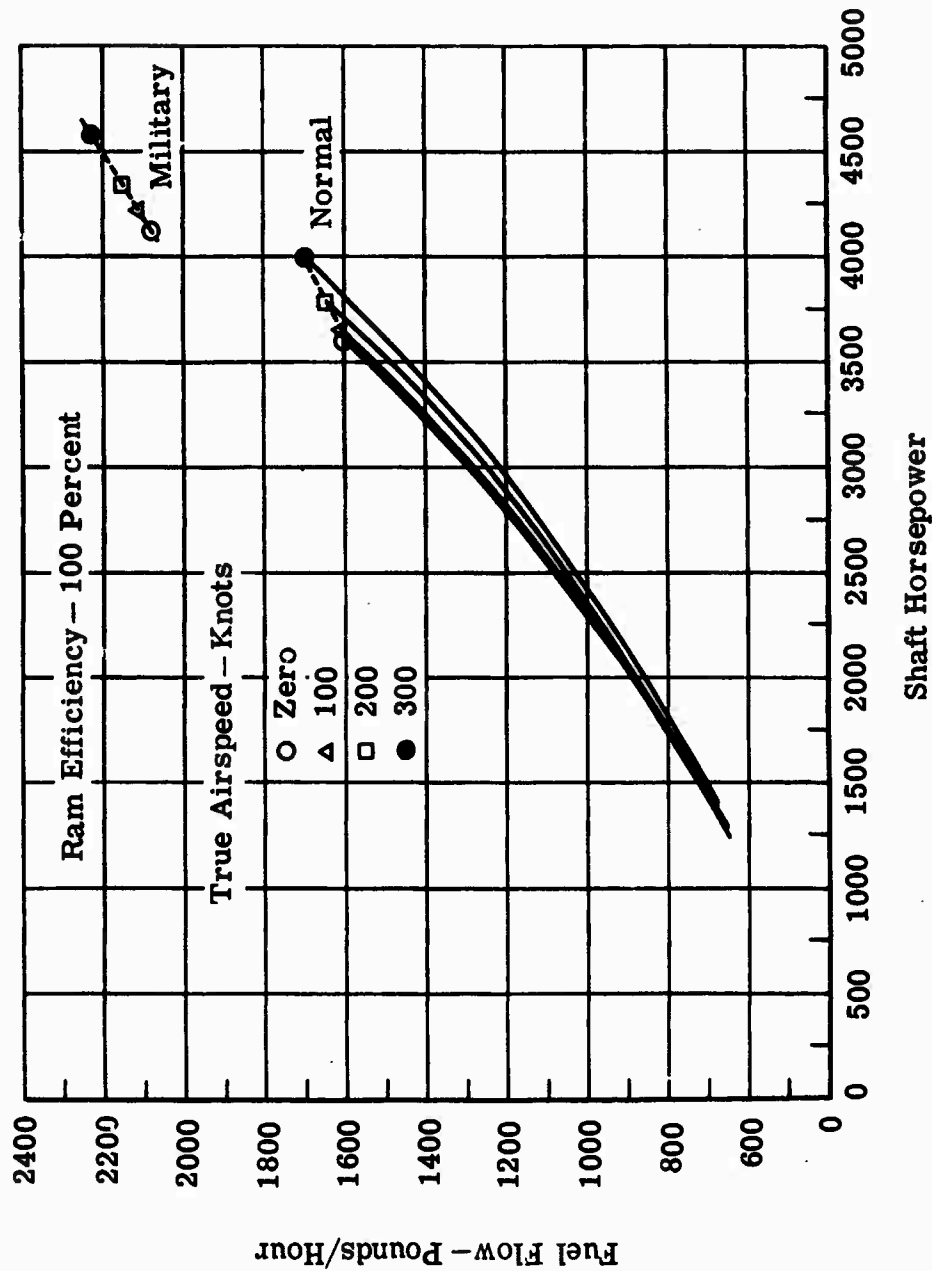
Figure 7. (C) Estimated Performance—Model 545-C3 (U)
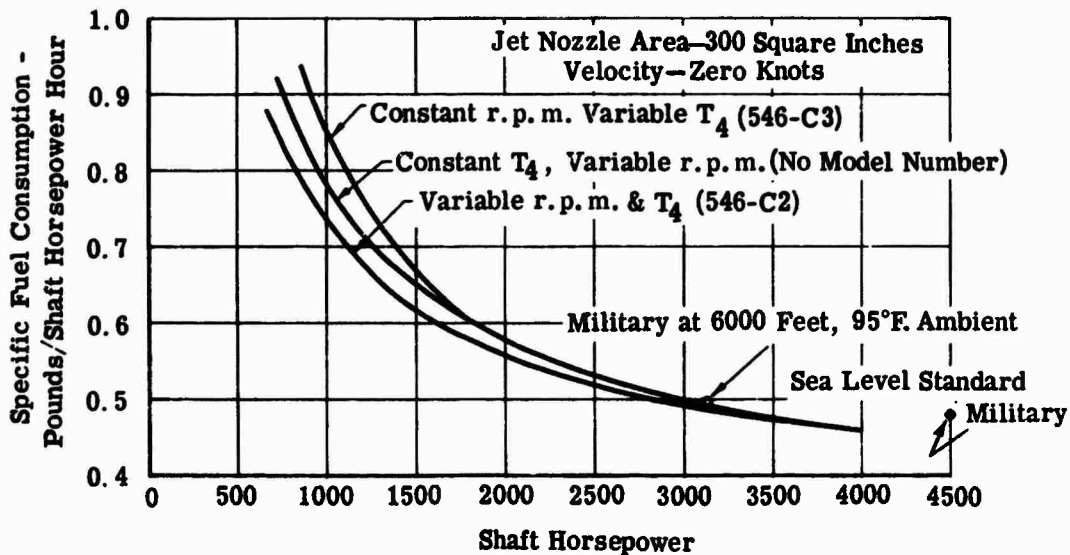
Figure 8. (C) Model 546 Nonregenerative Turboshaft
Engine Performance (U)

(U) Annular segmented admission was considered as a possible solu-
tion to the partial admission problem cited in the preceding paragraph.
Some investigators have reported that performance degradation need
not be incurred due to partial admission if care is exercised in closely
shrouding the turbine wheel in the "nonadmission" areas. Segmented
admission, however, approaches but does not duplicate partial admis-
sion.

(U) Windage losses can be a significant factor in the nonadmission
regions of a partial admission turbine. To effectively reduce these
losses with an engine out, some means must be found to closely shroud
the resulting inactive region of the wheel. This poses a difficult me-
chanical problem. Because of these problems and further mechanical
considerations, as indicated in the section titled Power Coupling Studies,
use of multiple turbines individually fed by a separate gas producer is
recommended.
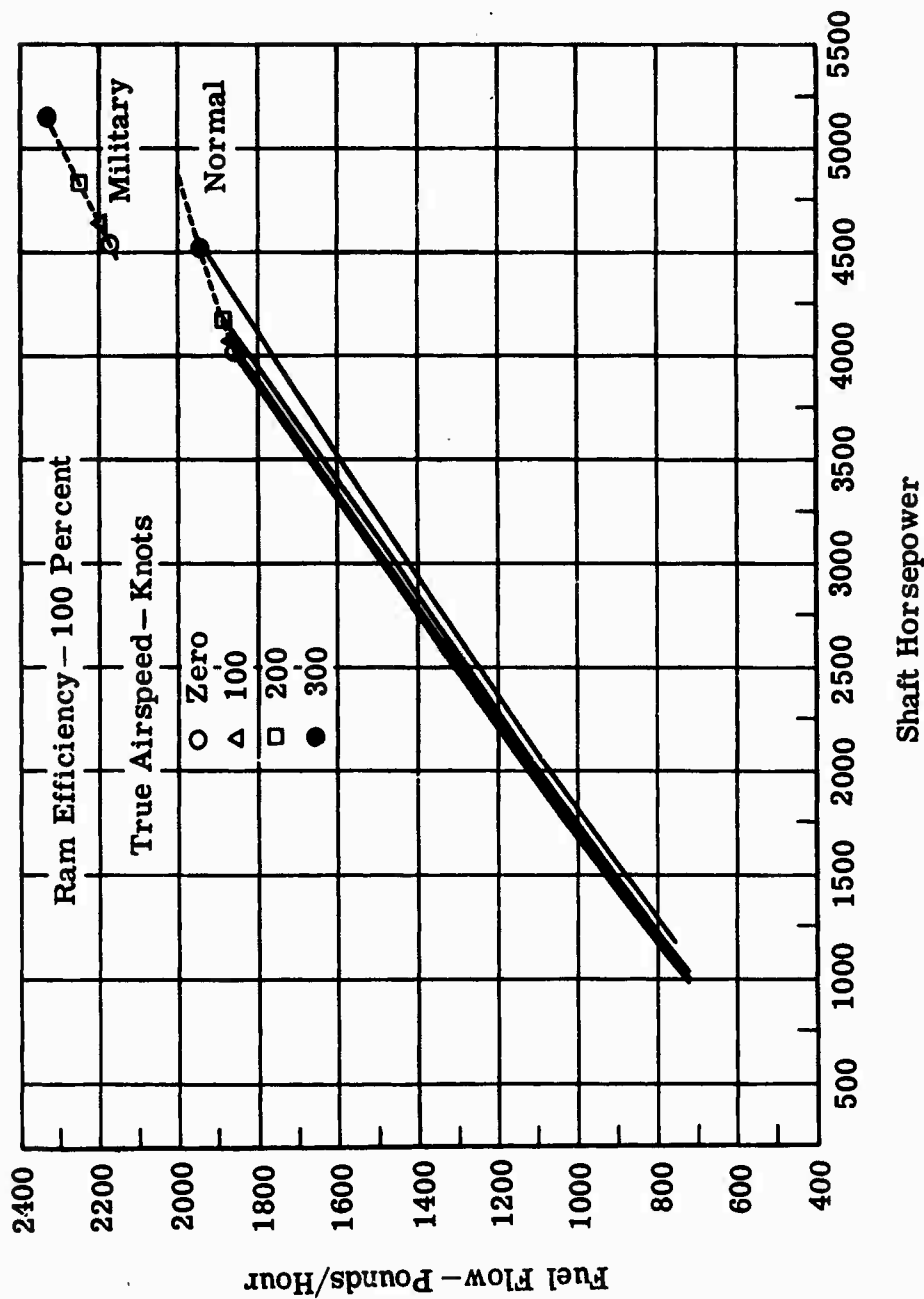
Figure 9. (C) Estimated Performance—Model 546-C2 (U)

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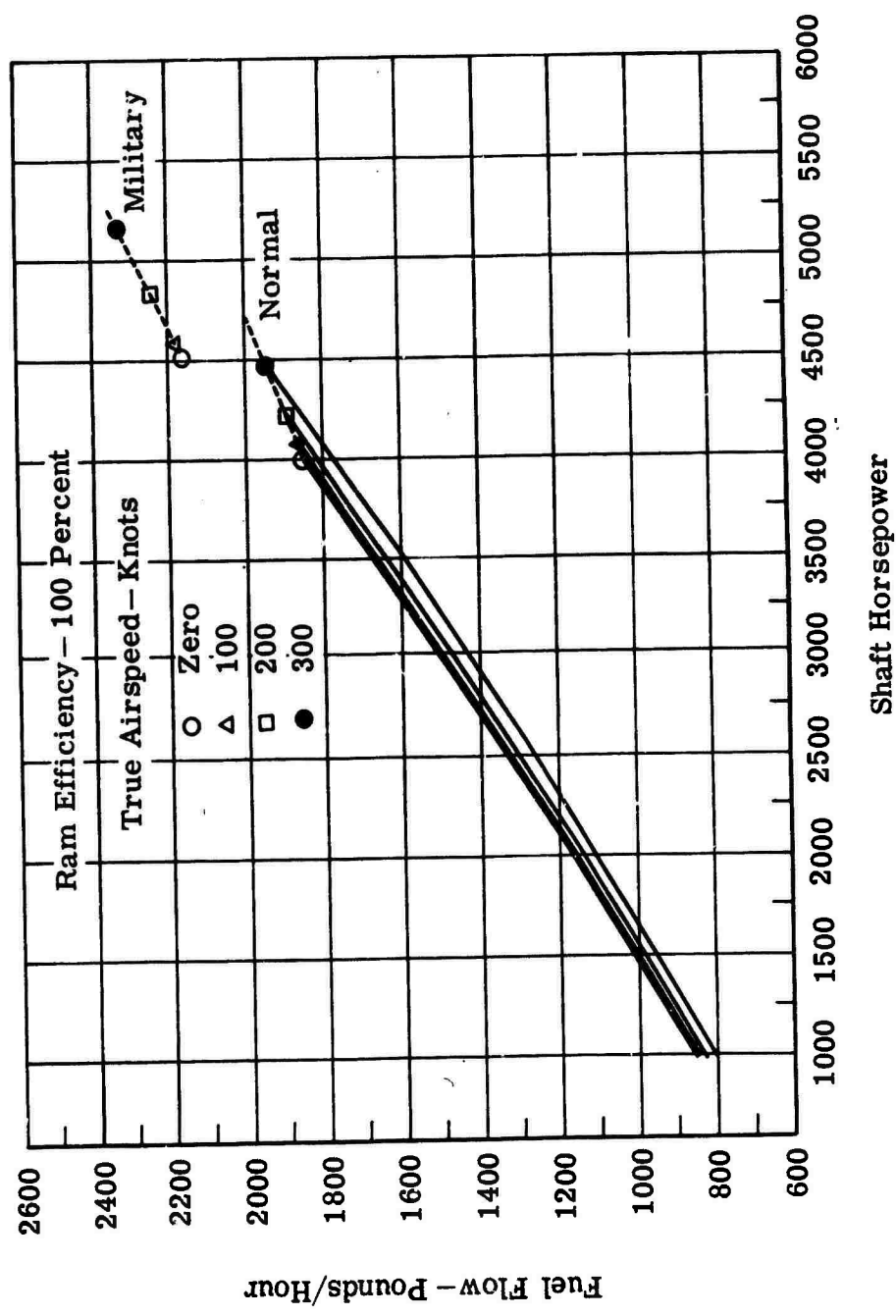
Figure 10. (C) Estimated Performance—Model 546-C3 (U)

22
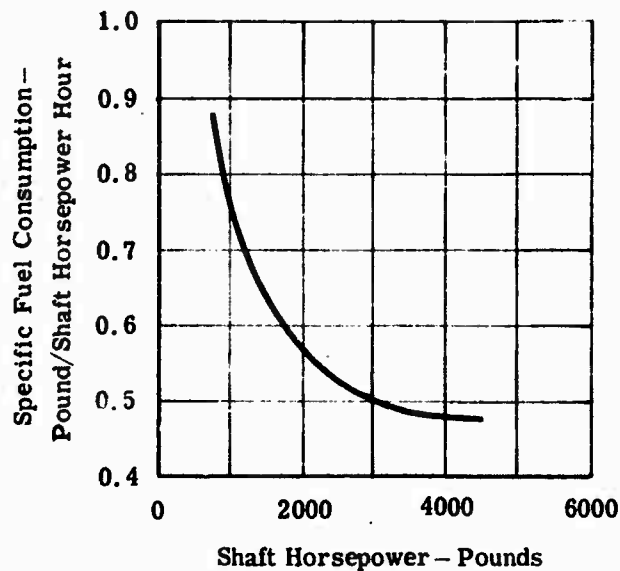
Figure 11. (C) Model 548-C2 Nonregenerative Free Turbine
Engine Performance (U)

(U) Relocation of a 548 power turbine from the axis of the gas producer
to a vertical position on the transmission will require ducting and turn-
ing the gas 90 degrees. The study indicates that a 4-percent pressure
loss can be expected in this duct. However, the resulting power loss is
offset by simplification in the mechanical elements of the drive system.

(U) The specific fuel consumption versus shaft horsepower relation-
ship for the 548 remote turbine (548-RT) drive system would be similar
to that of the basic 548-C2. The fuel consumption and shaft horsepower
for various true airspeeds for the gas-coupled drive approach are
presented in Figure 14.

(U) Variations of the T56 engine offer the maximum power available
for present or anticipated turboshaft engines. A fixed turbine version,
the Model 501-M25, would be capable of producing 6000 shaft horse-
power at standard sea level conditions. The specific fuel consumption-
shaft horsepower (SFC-SHP) characteristic of this engine is shown in
Figure 15. The effect of velocity on the performance of this engine is
shown on Figure 16.

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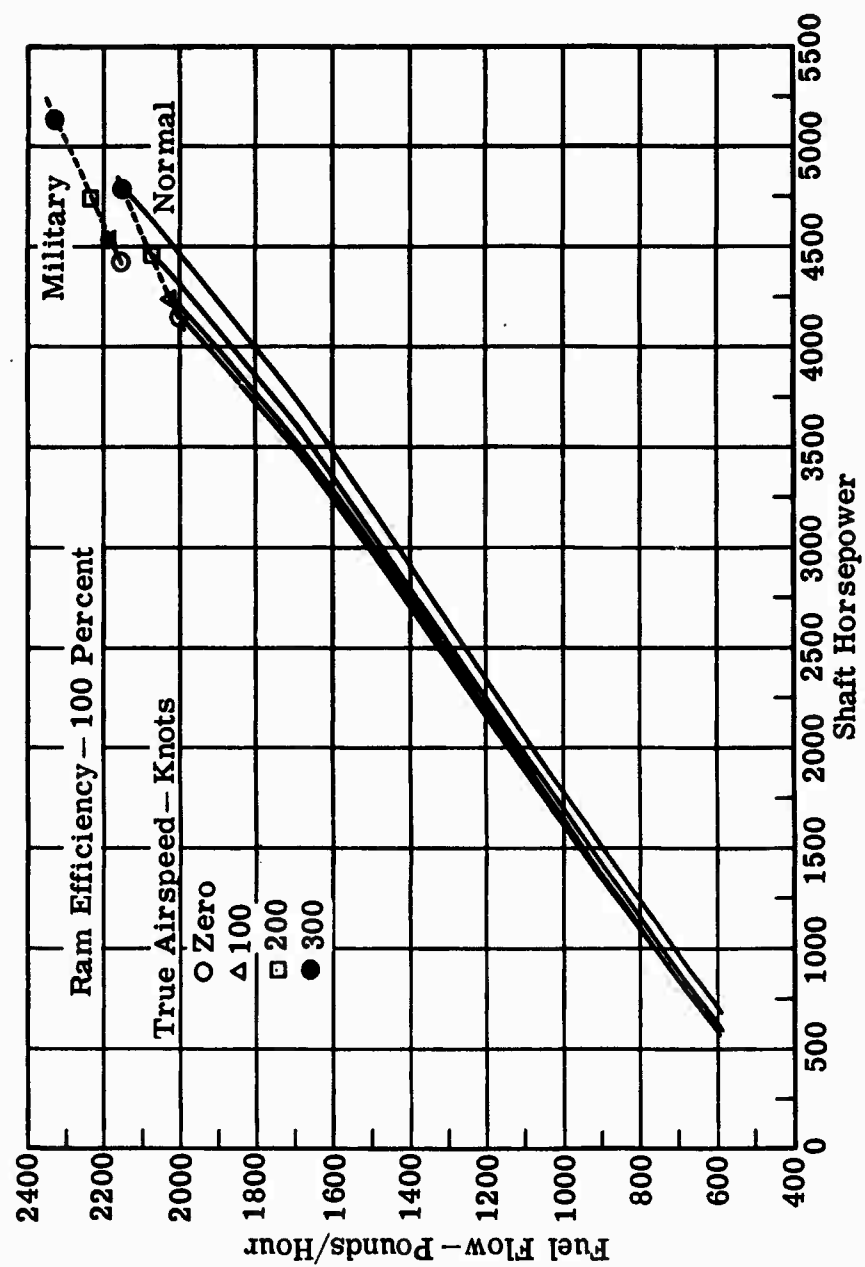
Figure 12. (C) Estimated Performance—Model 548-C2 (U)

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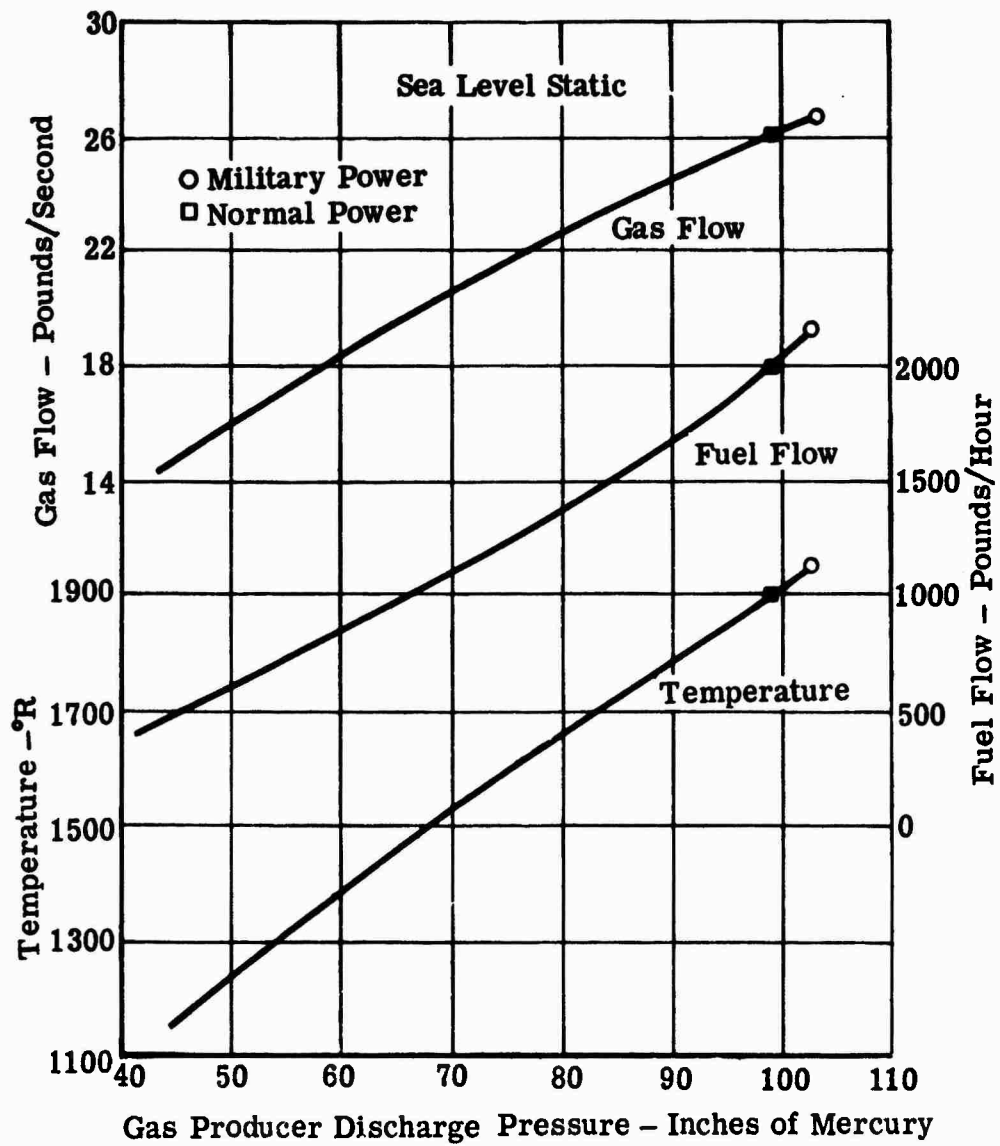
Figure 13. (C) Estimated Performance—Model 548-C2
Gas Producer (U)
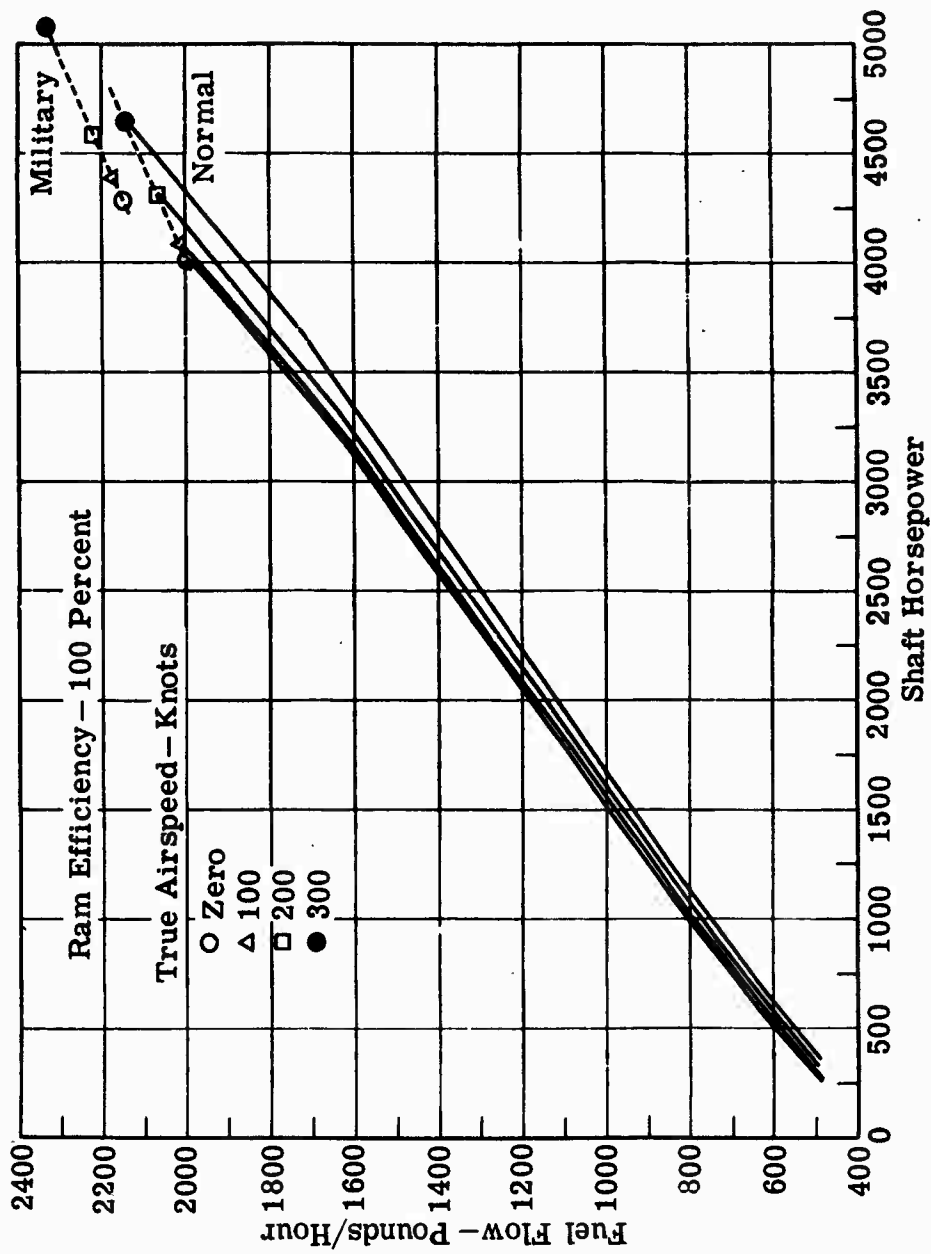
Figure 14. (C) Estimated Performance—Model 548-F.T (U)
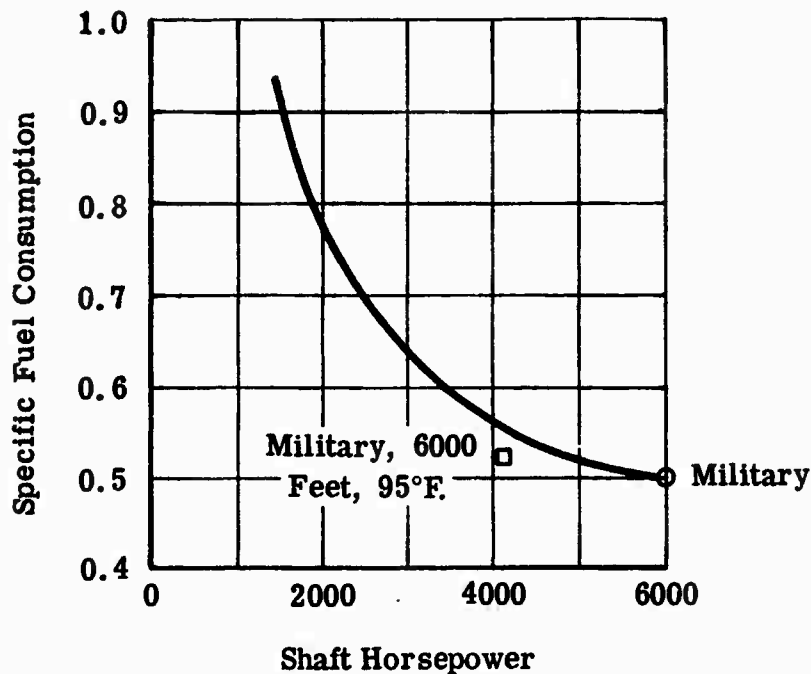
Figure 15. (U) Model 501-M25 Nonregenerative Turboshaft
Engine Performance

(U) A free turbine version of the T56, the Model 501-M26, would be capable of producing 5450 shaft horsepower and would exhibit the SFC-SHP characteristic shown in Figure 17. The effect of velocity on fuel consumption and shaft horsepower is shown in Figure 18.

(C) POWER AUGMENTATION (U)

(U) The maximum power requirement for the HLH is established by the 6000-foot, 95°F. transport-mission hover requirement as indicated in the subsection titled Mission Analysis. This requirement dictates the number of engines needed for the HLH, unless a suitable means of hot day power augmentation can be provided. The following methods of hot day power augmentation were considered:

● Water-alcohol injection
● Reheat between the gas producer and power turbines
● Hot-day ratings

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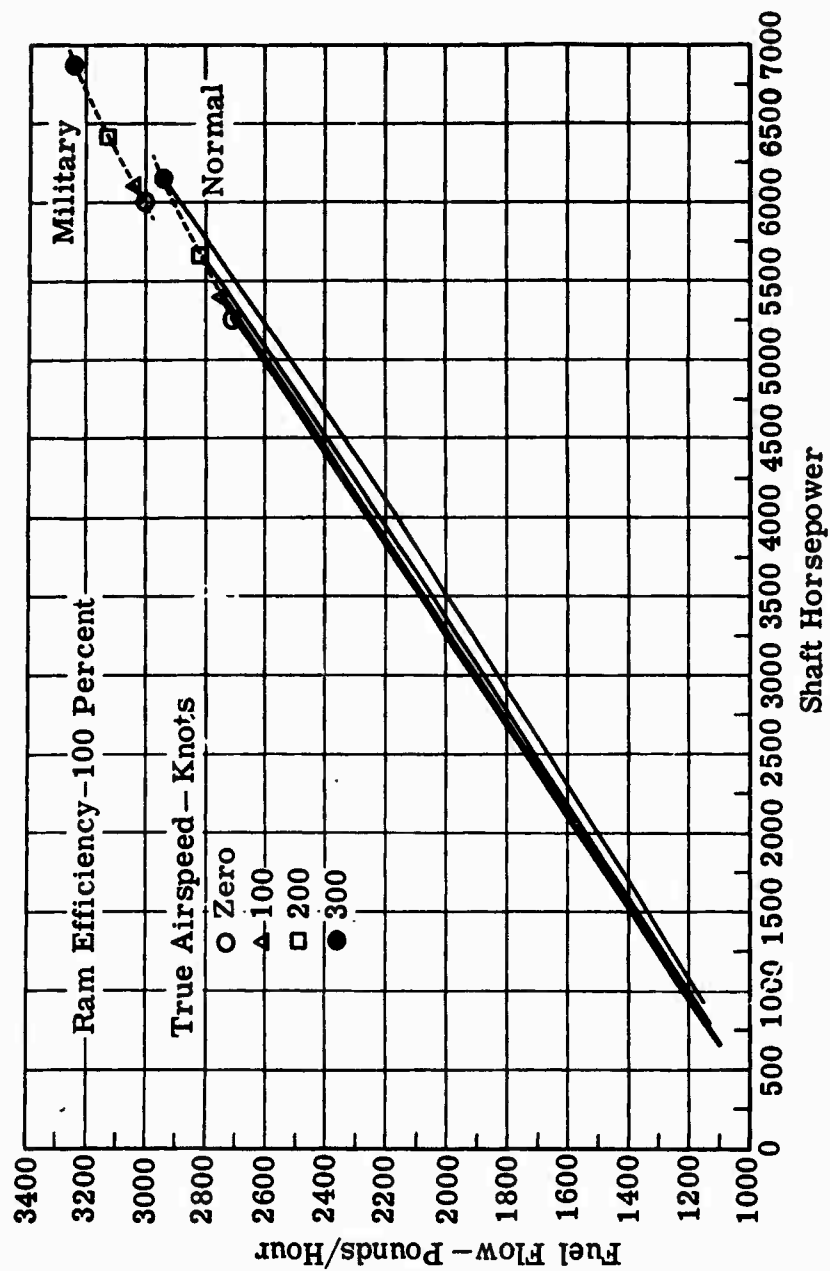
Figure 16. (U) Estimated Performance — Model 501-M25

28
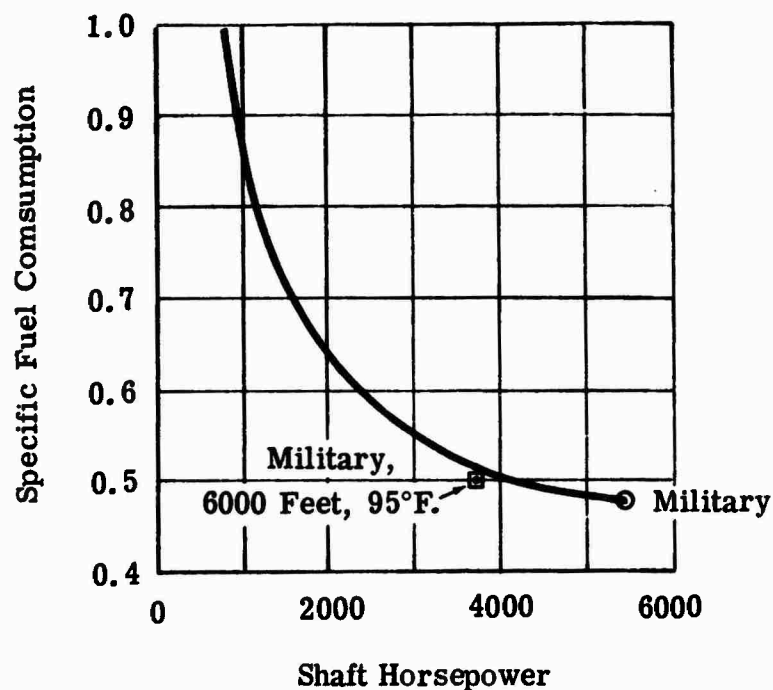
Figure 17. (U) Model 501-M26 Nonregenerative Free Turbine Engine Performance

(U) Water-alcohol injection has been extensively used to increase the thrust or power output of gas turbine engines on hot days. Injection of water-alcohol at the air inlet reduces the air temperature and results in an increase in air density and mass flow through the engine. In addition, the cooling effect of the water-alcohol within the compressor results in a reduction in compressor work per pound of inlet airflow. The combination of these effects results in an increase in useful turbine work.

(U) Water alone could be used as the injectant; however, unevaporated water can carry through the engine into the combustion chamber. This would require additional fuel and a more complex control. It has been found that if a mixture of two parts water to one part alcohol is used, fuel control modifications are not required. Furthermore, the use of alcohol alleviates the cold storage problem.

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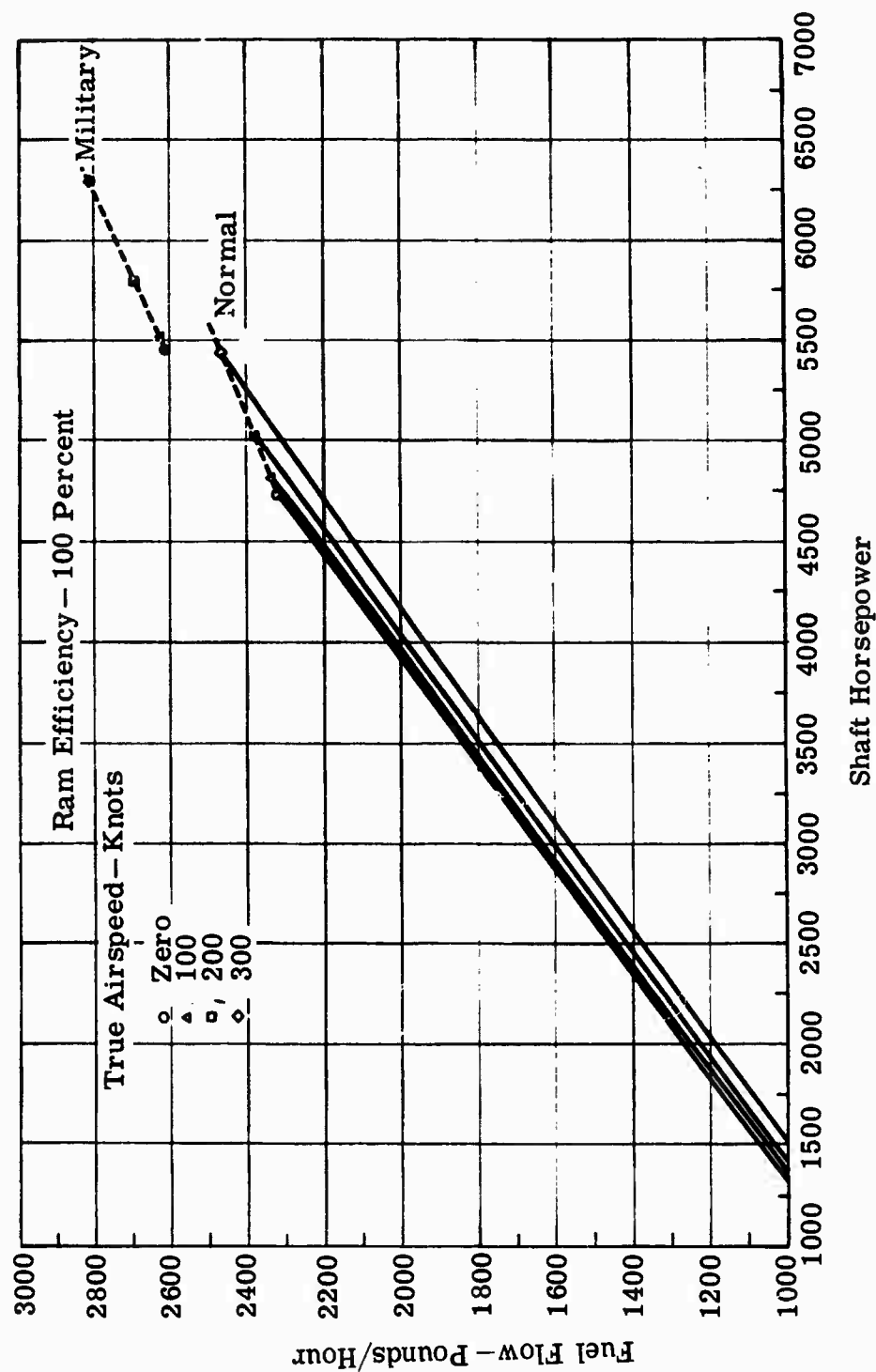
Figure 18. (U) Estimated Performance – Model 501-M26

(U) Water-alcohol augmentation is currently being used for the Model T56-A-10W engine which powers the Navy P3A. It is also used on the Model 501-D13H commercial engine. Based on experience with these engines, the percent of power augmentation versus injectant-airflow mass ratio which can be obtained on a 95°F. day is shown in Figure 19. It should be noted that the slope of this curve tends to become flat above 30-percent augmentation, indicating the limitation of this method.

(C) The considerations leading to determining the amount of water-alcohol required for the hot-day altitude hover condition, using three 548-C2 engines, are shown in Table 3. A total of 700 pounds of injectant would be required. The weight of the supporting systems (i.e., tank, pump, control, and plumbing) is dependent upon installation and reliability considerations. Experience with similar equipment for the
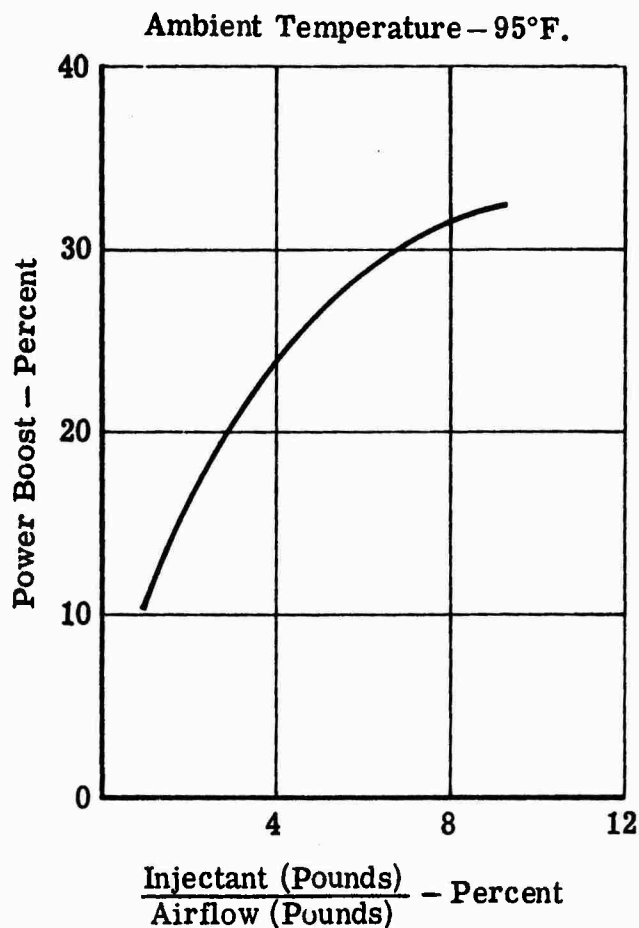
Figure 19. (U) Estimated Water-Alcohol Power Augmentation

T56-A-10W engine indicates that a hardware weight of 60 pounds per engine would seem reasonable. Therefore, the water-alcohol and supporting equipment weight for a three-engine 548-C2 powerplant would be 880 pounds. The lift-off weight with the water-alcohol system and fluid would be approximately 71,300 pounds. The lift-off weight of a comparable transport helicopter using four 548-C2 engines would be approximately 72,000 pounds, indicating a 700-pound savings if a three-engine, water-alcohol power augmentation system is used.

TABLE 3
(C) WATER-ALCOHOL POWER AUGMENTATION FOR THREE
MODEL 548-C2 ENGINES AT 6000 FEET, 95°F. (U)

	Initial Hover	Midpoint Hover
Gross Weight—pounds	70,400	67,000
Power Required—shaft horsepower	11,100	10,400
Power Available—shaft horsepower	8,775	·8,775
Required Augmentation—percent	26.5	18.5
Water-Alcohol/Airflow Mass Ratio	0.05	0.025
Airflow—pounds per second	19.5	19.5
Hover Duration—minutes	3	2
Water-Alcohol Required—pounds per engine	175 .	58
Water-Alcohol Required—pounds per hover	525	174
Total Water-Alcohol Required—pounds		699

(U) The major disadvantage in employing water-alcohol for power augmentation is that de-ionized water is required. Use of untreated water will result in compressor contamination and performance degradation. The desirability of using this method of augmentation depends on the trade-off between the logistic problem and the reduced weight and reduced maintenance requirement for a three-engine powerplant.

(U) Power augmentation using reheat in the turbine section was considered. This could be accomplished with a free-turbine engine by installing a diffuser and combustion section between the gas producer and power turbines. This method of augmentation would utilize engine fuel and eliminate the logistics problems associated with the water-alochol approach.

(C) A performance analysis was made using Model 548 characteristics to determine how much power could be gained with reheat and the attendant performance penalties which would be incurred during normal operation. As shown in Figure 20, a maximum net augmentation of 20 percent can be obtained if reheat is used to increase the power turbine inlet temperature to 2060°F. Comparison of the "with reheat" and "without reheat" curves shows an 8-percent power loss due to the reheat diffuser and combustor pressure losses during normal operation. These losses, as indicated by the curves, are reflected in the output and specific fuel consumption throughout the operational range of the engine.
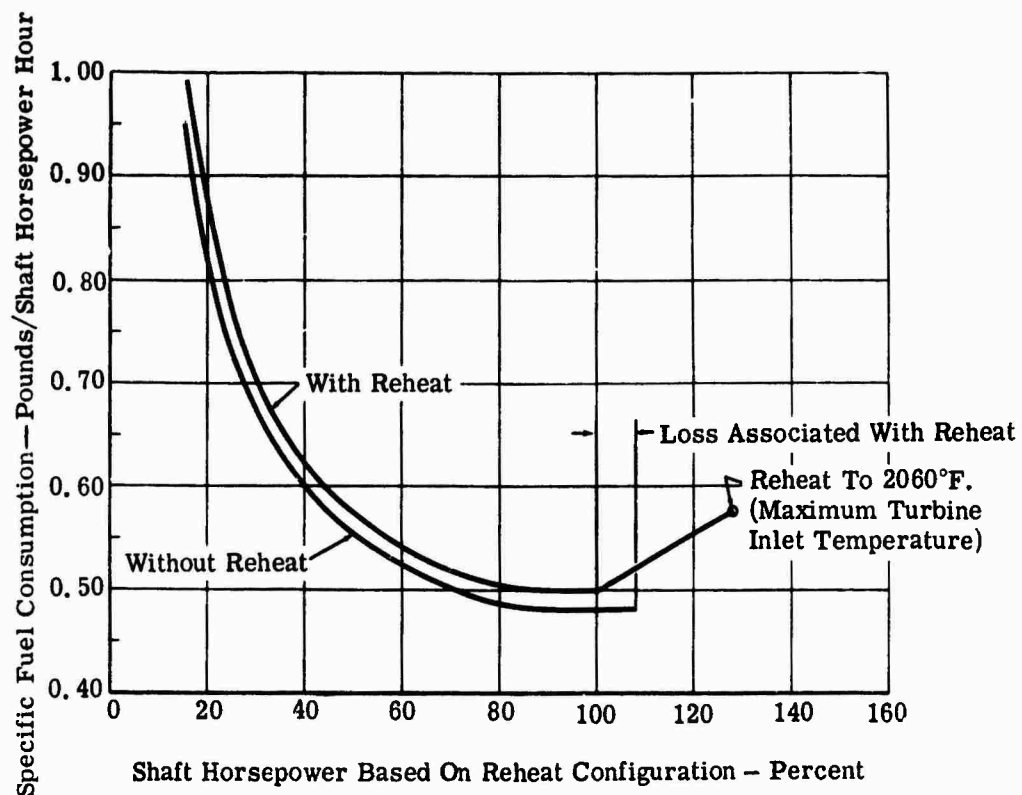
Figure 20. (C) Reheat Power Augmentation—Model 548-C2 (U)

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(U) To accommodate lower density gas during the reheat operation, the power turbine inlet area must be increased by 17 percent. Therefore, variable inlet geometry (with its attendant development problem) must be provided. In addition, approximately 80 pounds of hardware are required per engine, and cooling air must be provided for the power turbine blades and vanes. Because of the relatively low power gain and the attendant performance and mechanical problems, reheat is not recommended as a means of power augmentation.

(U) Another method of increasing power for hot-day operation is to increase the turbine inlet temperature and engine speed. Figure 21 shows the effect of increased engine speed and turbine inlet temperature on the shaft power of a Model 548 engine operating at 6000 feet and at 95°F. ambient. As shown in Table 3, 26.5-percent power augmentation is needed to meet the altitude hot-day hover requirement of the transport mission using three Model 548 engines. As indicated in Figure 21, this degree of augmentation would require increasing the turbine inlet temperature by 170°F. and the engine speed to 104 percent of design speed. This increase in operating limits may be attainable in the 1970 time period.

(U) As indicated in a subsequent section titled Number of Engines, the use of three 501-M26 engines would require approximately 8-percent augmentation and the use of four 545-C2 engines would require approximately 4.4-percent augmentation. This degree of augmentation is readily attainable by each of the three methods discussed.

(C) REGENERATOR ANALYSIS (U)

(U) The regenerator for the Model 545 engine is a high-effectiveness, low-pressure-drop, lightweight, compact heat exchanger with an exhaust bypass valve controlled by the power lever.

(U) A fixed, tubular-type heat exchanger was selected for the Model 545 engine based on Allison experience in the design and testing of the following:

- Full-scale 5500 equivalent shaft horsepower fixed tubular regenerator engine .
- Full-scale 4100 equivalent shaft horsepower rotating regenerator
- Subscale fin-plate-type regenerator
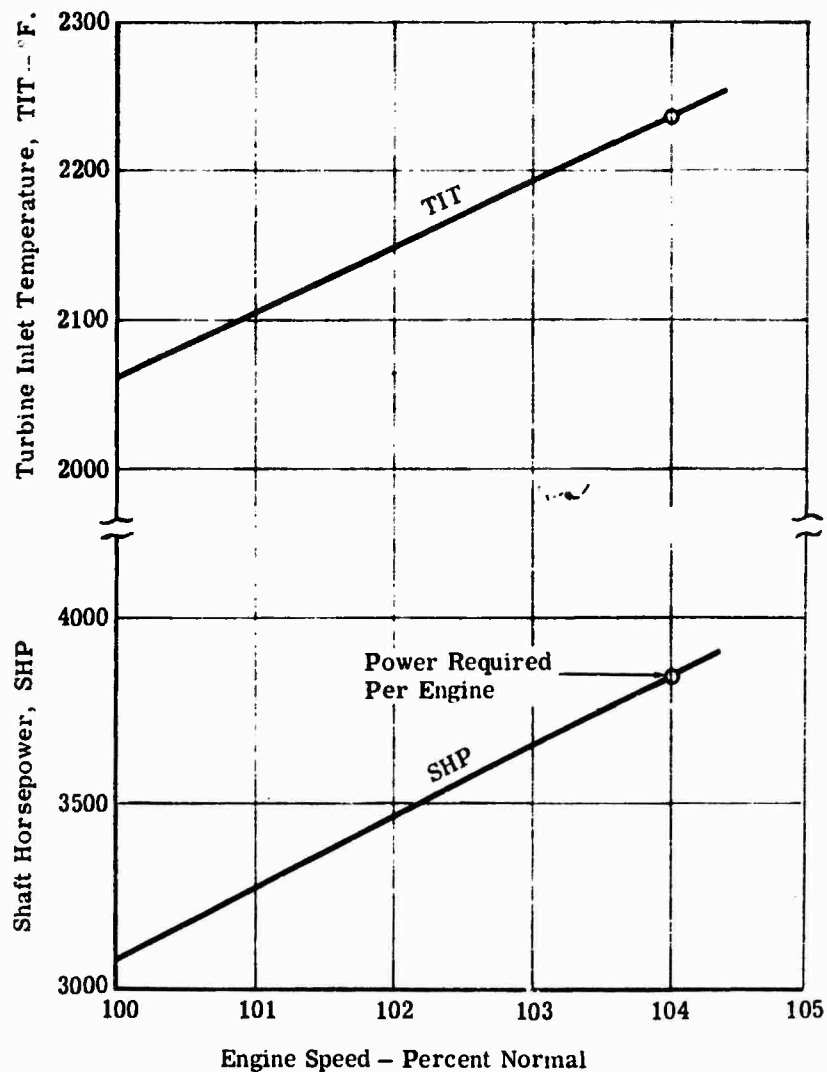
Figure 21. (C) Power Augmentation With Increased Turbine Inlet
Temperature and r. p. m. —Model 548-C2 (U)

(U) A fixed, tubular-type regenerator, as shown in Figure 22, was
selected for the Model 545 because indications were that the rotating
regenerator could not meet the endurance and reliability requirements
in the specified time. Also, the fin plate was considered to be heavier,
to be more difficult to fabricate, and to have higher thermal stresses than
the tube type.

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Air From Compressor

Turbine Exhaust

Bypass Valve

Air to Combustor
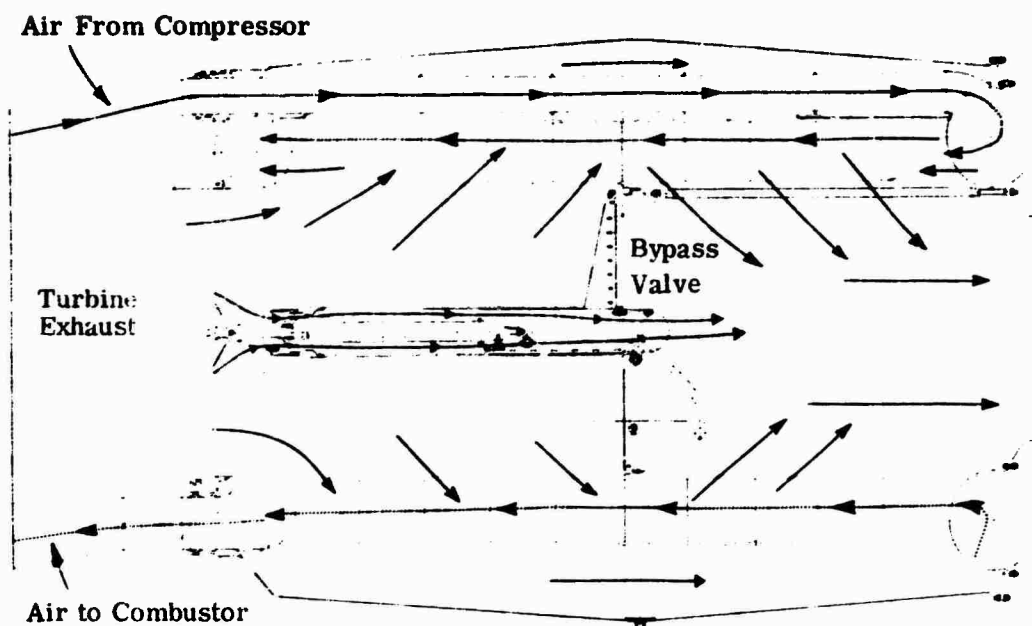
Figure 22. (U) Model 545 Regenerator Flow Path

(U) The basic heat transfer data for the tube-type regenerator were based on experimental and analytical data. The effectiveness and pressure drop of the Model 545 regenerator were optimized for the missions specified by the U. S. Navy for the Model T78-A-2 regenerative engine. The resulting design is not necessarily the optimum for other missions and other applications. Therefore, it was necessary to resize the present configuration to determine the optimum configuration for the various HLH missions. For the optimization study, the inside diameter of the regenerator was held constant so that redesign of the turbine diffuser would not be necessary for mating the engine and regenerator. Regenerator weight was determined over a range of effectiveness and pressure drop. Figure 23 shows the resulting relationship between effectiveness, pressure drop, and core weight for the tubular-type design. Core weight only is shown in Figure 23 because it is amenable to analytical procedures. A factor of 2.4 times core weight, based on Model 545 experience, was used in the optimization analysis to determine total regenerator weight.

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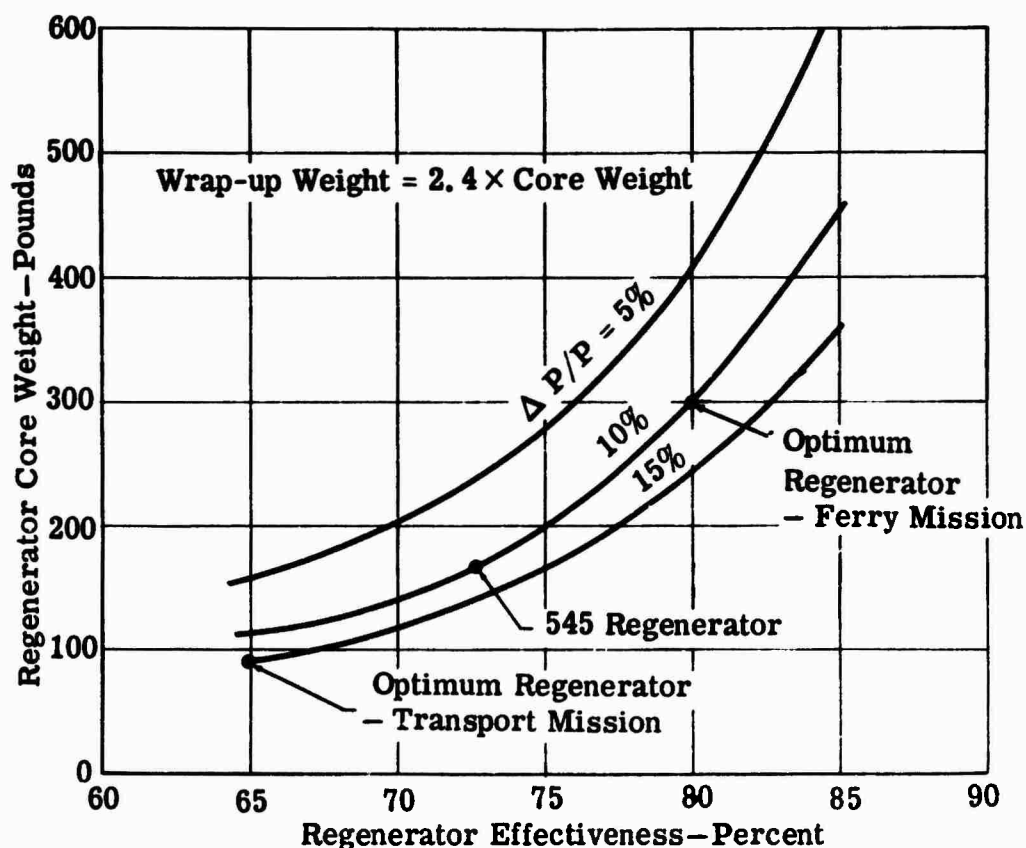
Figure 23. (U) Tubular-Type Regenerator Characteristics

(U) Gross weight and fuel consumption were computed (Figures 24, 25, and 26) for each mission over a range of effectiveness and pressure drop. Use of regenerative engines would result in a reduction in lift-off weight and fuel used or an extension of range for the ferry mission. The range of the heavy-lift mission is such that use of regeneration does not show any reduction in gross weight. The transport mission would benefit only slightly with some regeneration. However, fuel can be saved on all missions if a regenerator is used. The magnitude of this savings is shown in the Mission Analysis section of this report.

(U) As indicated in Figures 24, 25, and 26, there is an optimum regenerator for each mission. Thus, one regenerator represents a compromise for some missions. The ferry mission needs a regenerator

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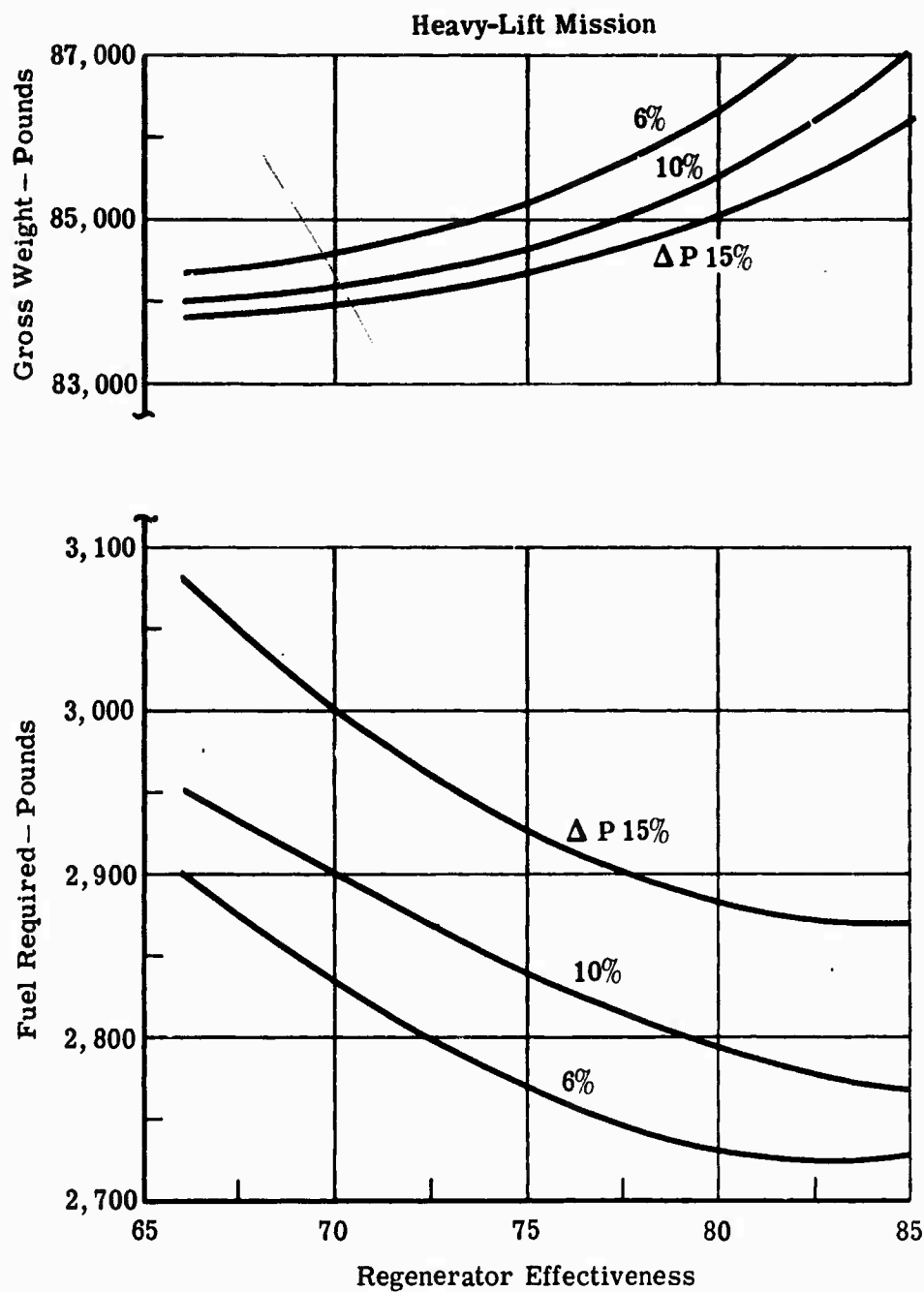
Figure 24. (C) Regenerator Optimization—Heavy-Lift Mission (U)

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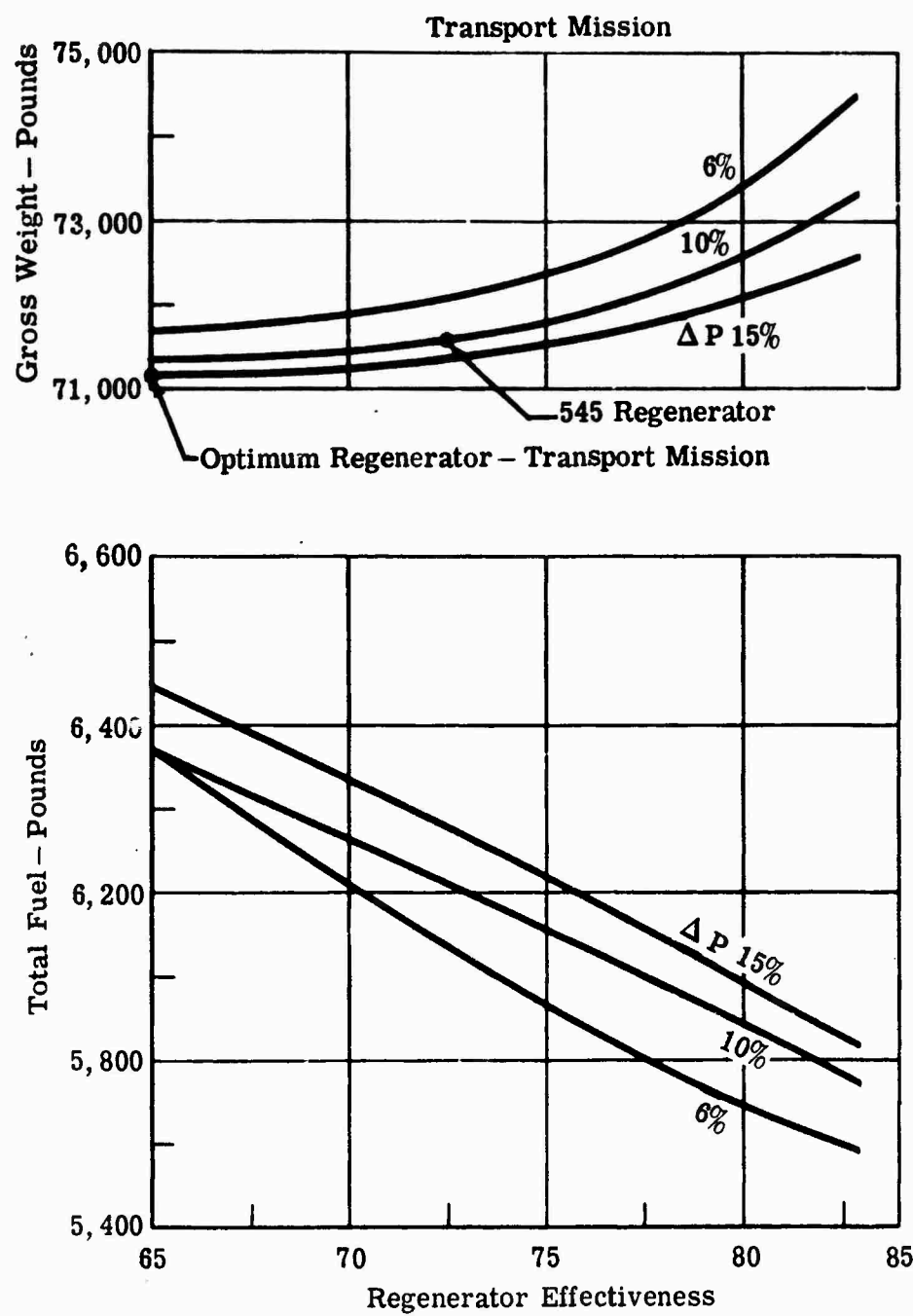
Figure 25. (C) Regenerator Optimization—Transport Mission (U)
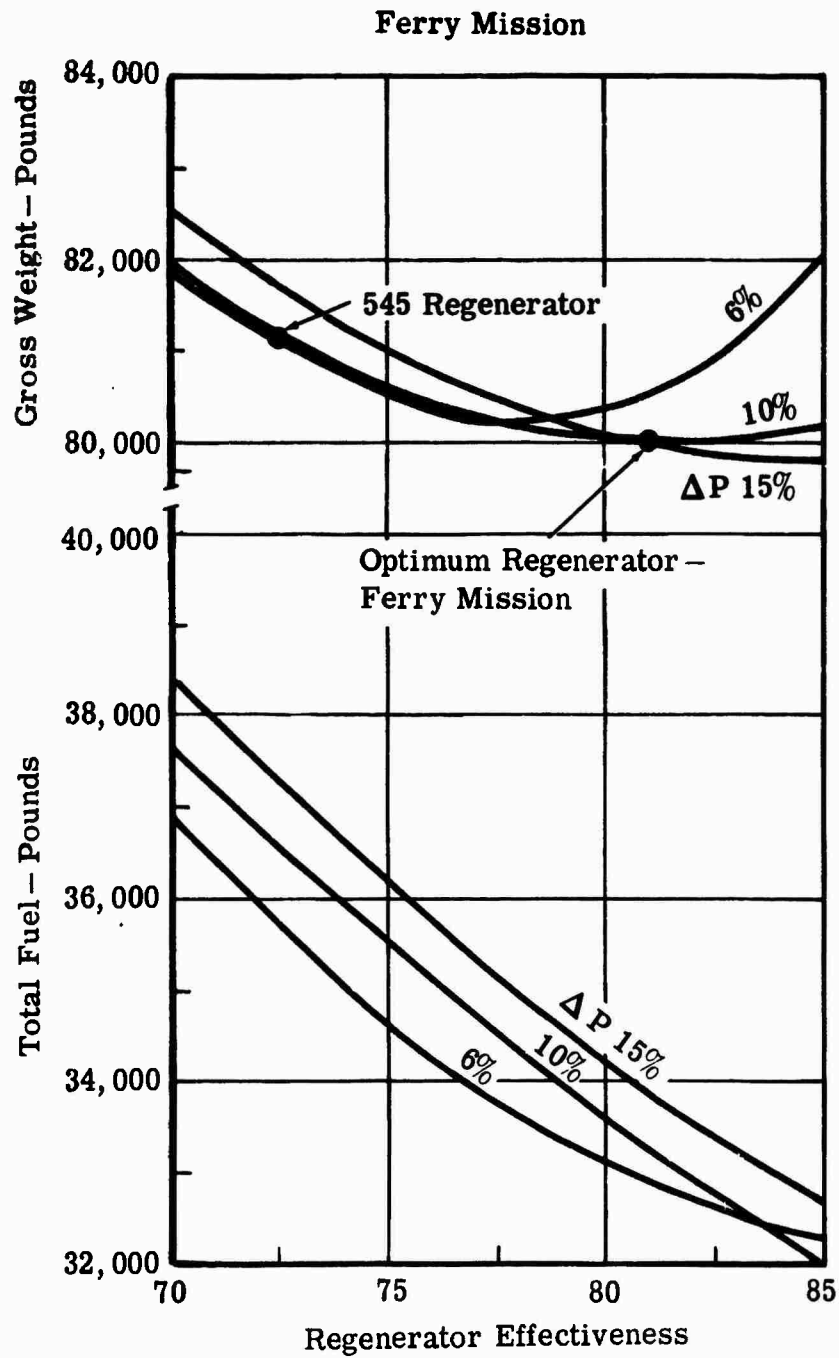
Figure 26.   (C)   Regenerator Optimization—Ferry Mission (U)

with a higher effectiveness than that of the Model 545 and the transport would optimize at a lower effectiveness. Therefore, the Model 545 would be a reasonable compromise.

(U) Table 4 compares the approximate dimensions of the optimum regenerators for each mission with that of the Model 545.

TABLE 4
(U) COMPARISON OF OPTIMUM REGENERATORS AND
MODEL 545 PARAMETERS

	Model 545	Ferry	Transport
Outside Diameter—inches	35	34.7	31.4
Inside Diameter—inches	19.2	19.5	19.5
Length—inches	56	68	47
Core Weight—pounds	150	300	90
Total Weight—pounds	376	750	225
Effectiveness—percent	72.6	80	65
ΔP—percent	10	10	15

(U) Regenerator dimensions and weight given in Table 4 were obtained from a computer design program used in sizing the Model 545. It is necessary to select some engine power setting from which to compare the regenerators. The effectiveness and pressure drops presented in Table 4 occur at 92.5-percent engine speed or approximately 75-percent normal power setting.

(U) A regenerative free turbine implies variable turbine geometry to obtain full benefits from the regenerator. If the Model 548-C2 were to be equipped with variable power turbine geometry and a regenerator, its performance would be approximately equal to the Model 545.

(C) MISSION ANALYSIS (U)

(U) The objectives of the mission studies were to determine the number of engines required, the helicopter gross weight, and the fuel consumed for specified missions using various engine models. The effect of in-flight shutdown of engines and the use of hot-day power augmentation were also determined. The missions defined by USAAML are given in Table 5.

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TABLE 5
(U) HLH MISSIONS

	Transport	Heavy Lift	Ferry	Special Heavy Lift
Payload (Outbound), tons	12	20	0	16
Distance, nautical miles	**100 (radius)**	**20 (radius)**	**1500 (range)**	**100 (radius)**
Cruise Velocity (With Payload), knots	110	95		95
Cruise Velocity (Without Payload), knots	130	130	Best for Range	130
Cruise Altitude	SLS	SLS	SLS	SLS
Hover Capability (OGE)	6000 feet, 95°F.	SLS	—	SLS
Hovering Time, minutes	3 at takeoff	5 at takeoff	—	5 at takeoff
Notes:	2 at midpoint (with payload)	10 at destination (with payload)		10 at destination (with payload)

Reserve Fuel—10 percent of initial fuel for all missions
Fuel Allowance—for start, warm-up, and takeoff per MIL-C-5011A for all missions
SLS—Sea Level Standard
OGE—Out of Ground Effect

(U) Method of Analysis

(U) Three primary and interacting factors were involved in the mission analysis: the gross weight of the helicopter; the power required from the engines to support this weight; and the fuel consumed in providing this power. These factors vary with time and operational mode. Because of the interacting nature of these parameters, an iterative process was used. This process is described in detail in the appendix—Mission Analysis Method; therefore, only the basic procedures are discussed in the following paragraphs.

(U) Since a number of engine models and configurations were considered for the various missions, it was necessary to establish a baseline weight for an empty helicopter, less engines. The following assumptions were used in establishing this baseline value:

● Empty weight/design gross weight for the heavy-lift mission— 0.47
● Installed engine weight equals 1.4 times the bare engine weight
● Specific fuel consumption penalty—5 percent
● Installation power loss—5 percent
● Crew and trapped fluid weight—700 pounds
● Fuel reserve—10 percent
● Additional tankage weight for ferry mission—7.5 percent of the difference in fuel weight between transport and ferry missions

(U) Using the assumptions noted, the lift-off weight was established for a heavy-lift mission helicopter powered by four Model 548-C2 free-turbine engines. Once the lift-off weight was established by the iterative process described in the appendix, the baseline empty weight (less engines) was obtained by subtracting the fuel, payload, installed engine, and crew and trapped fluid weights. Using this procedure the empty weight (less engines) was found to be approximately 34,700 pounds, and this value was used for all missions and engine configurations considered.

(C) Number of Engines (U)

(U) The number of engines required for the HLH is determined by the transport mission, 6000-foot, 95°F. ambient hover requirement. Relative to sea level standard day performance, the power required to lift a given load is increased and the power available from the engines is decreased at the altitude hot-day conditions. The power required versus gross weight for these conditions is shown in Figure 27. This relationship was determined from the sea level standard-day static power requirements established by USAAML (see Figure 1) by applying a density correction to power and to gross weight.

(U) The effect of altitude and ambient temperature on engine shaft horsepower is shown in Figure 28. The shaft horsepower attainable at 6000-foot, 95°F. ambient temperature is approximately 68 percent of that attainable at sea level standard-day conditions. This power degradation is typical of gas turbine engines, unless hot-day ratings or power augmentation is employed.

(U) The power available versus number of engines for the various models studied are shown in Figures 29 through 33. The power available from each engine was reduced by 5 percent from that shown previously in the Performance Analysis subsection, to allow for installation losses. Three lateral lines appear on each figure. The upper and lower lateral lines show the power required for the transport altitude hot-day hover for helicopters with heavy-lift design gross weights of 85,000 and 75,000 pounds, respectively. The lift-off weight for the transport mission varies with each engine model and is not called out on the curves. The dashed lateral line shows the power required for a helicopter which has an empty weight (less engines) of 34,700 pounds. It is believed that this weight is representative of what could be expected with shaft-driven heavy-lift helicopters in the 1970 time period.

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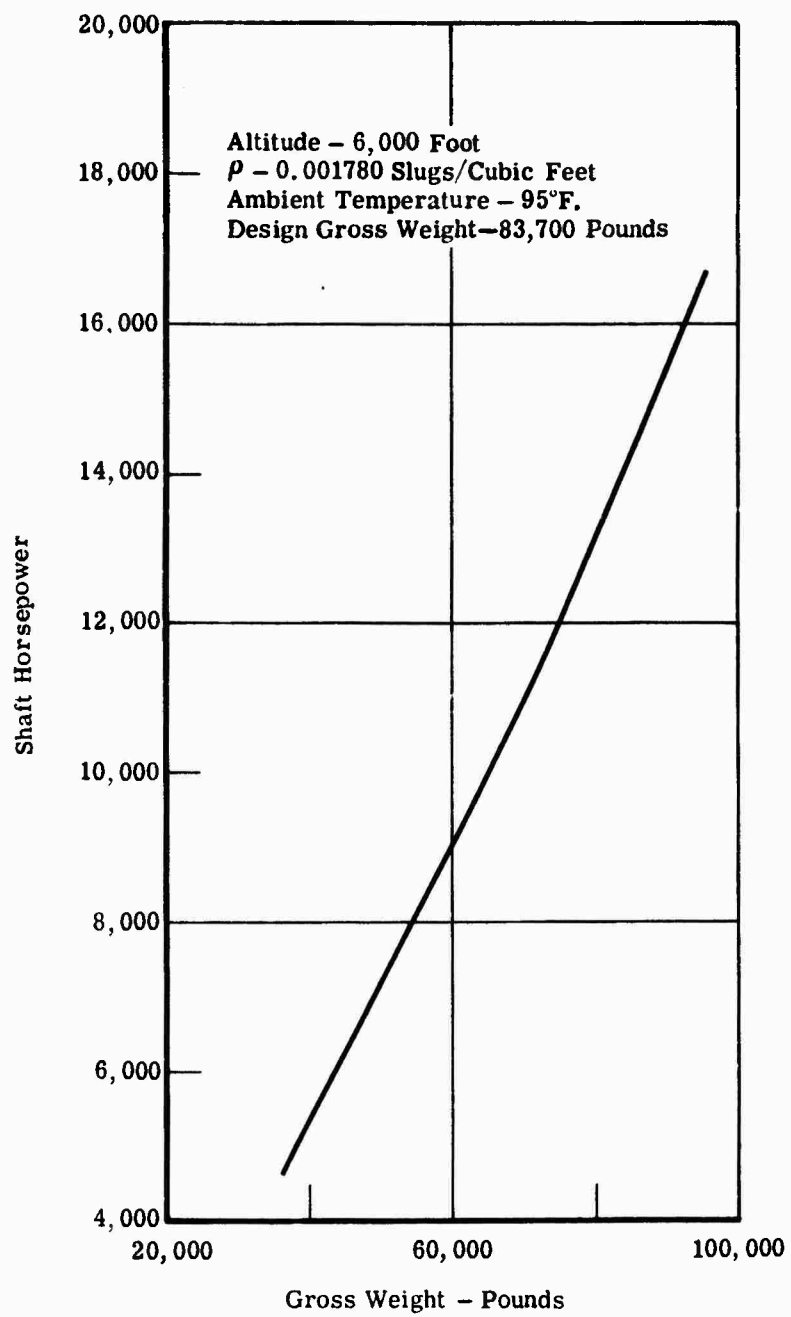
Figure 27. (U) Power Required to Hover for 6,000-Feet,
95°F. Ambient

44
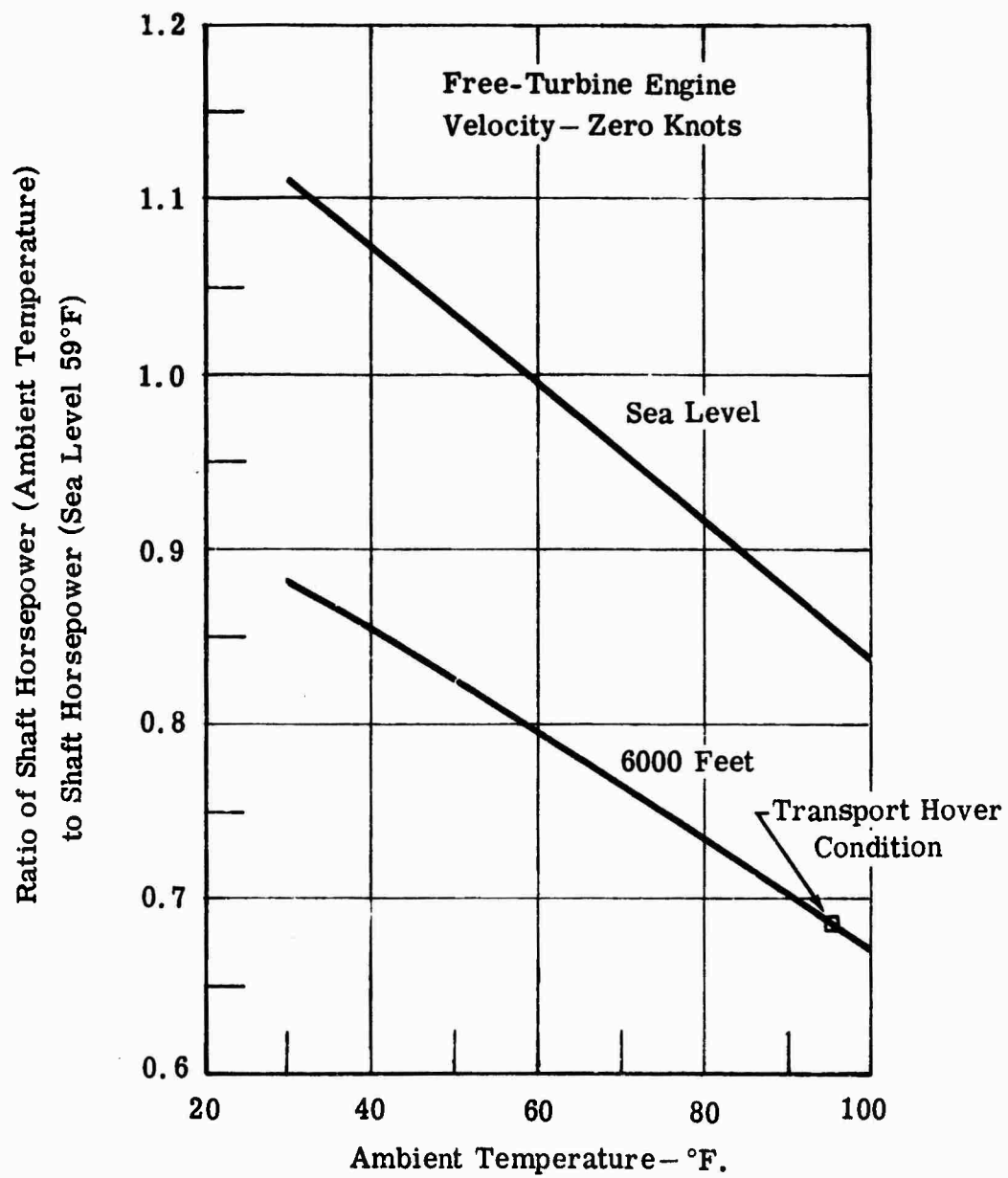
Figure 28. (U) Altitude and Temperature Effect on Power

45

Altitude – 6000 Feet
Ambient Temperature – 90°F.
Installation Loss – 5%
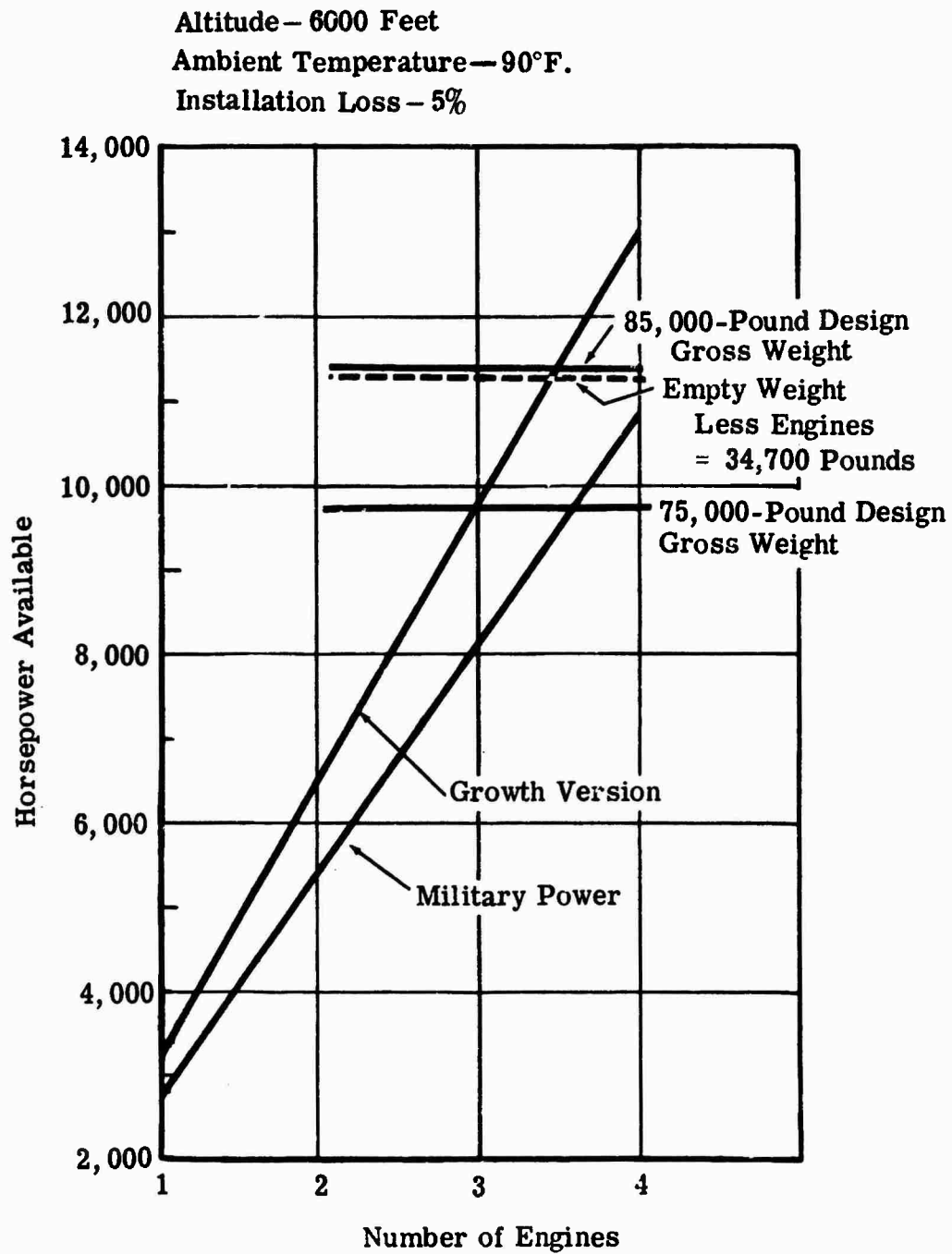
Figure 29. (C) Number of Engines Required—Models
545-C2 and 545-C3 (U)

46
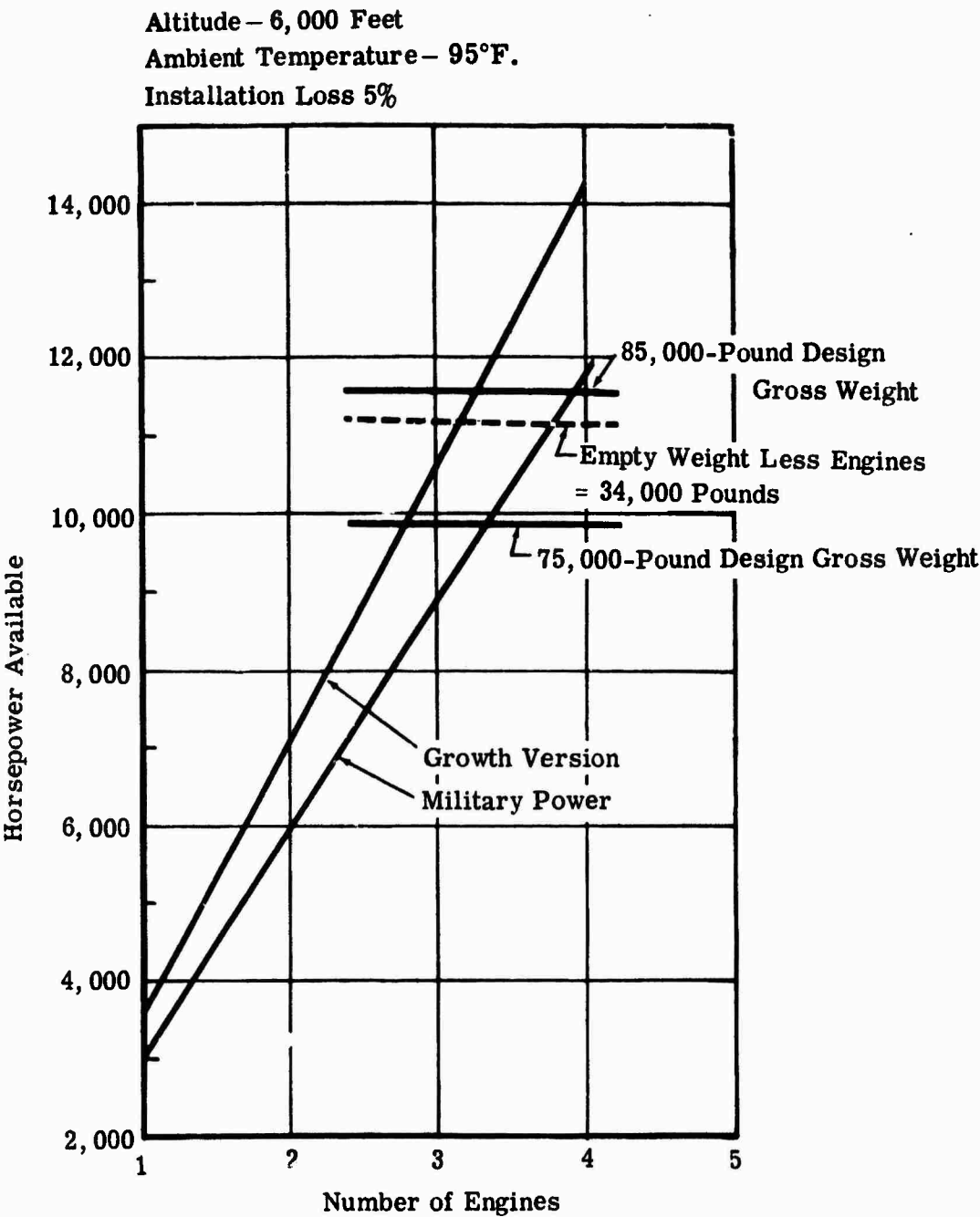
Figure 30. (C) Number of Engines Required—Models
546-C2 and 546-C3 (U)
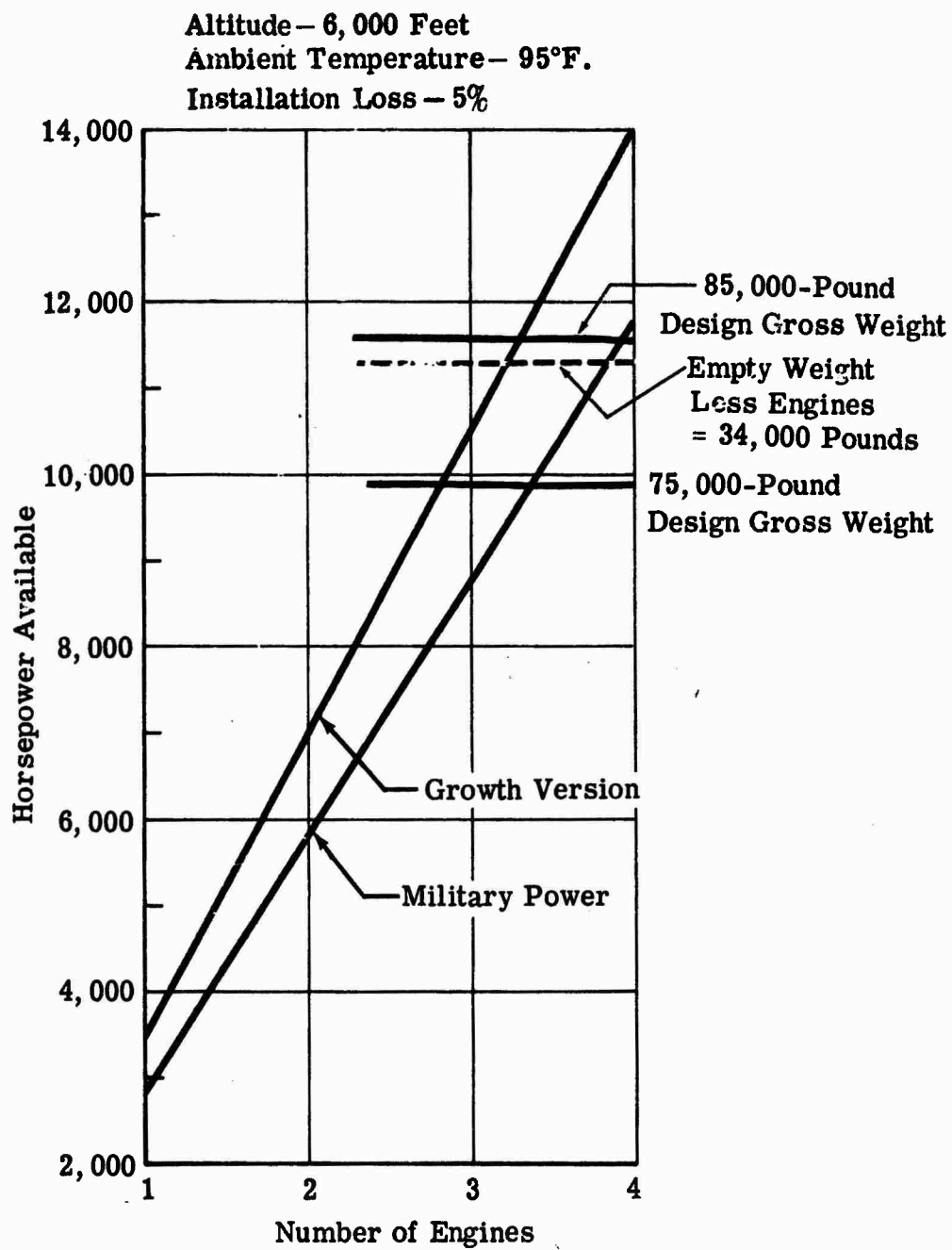
Figure 31. (C) Number of Engines Required—Model 548-C2 (U)
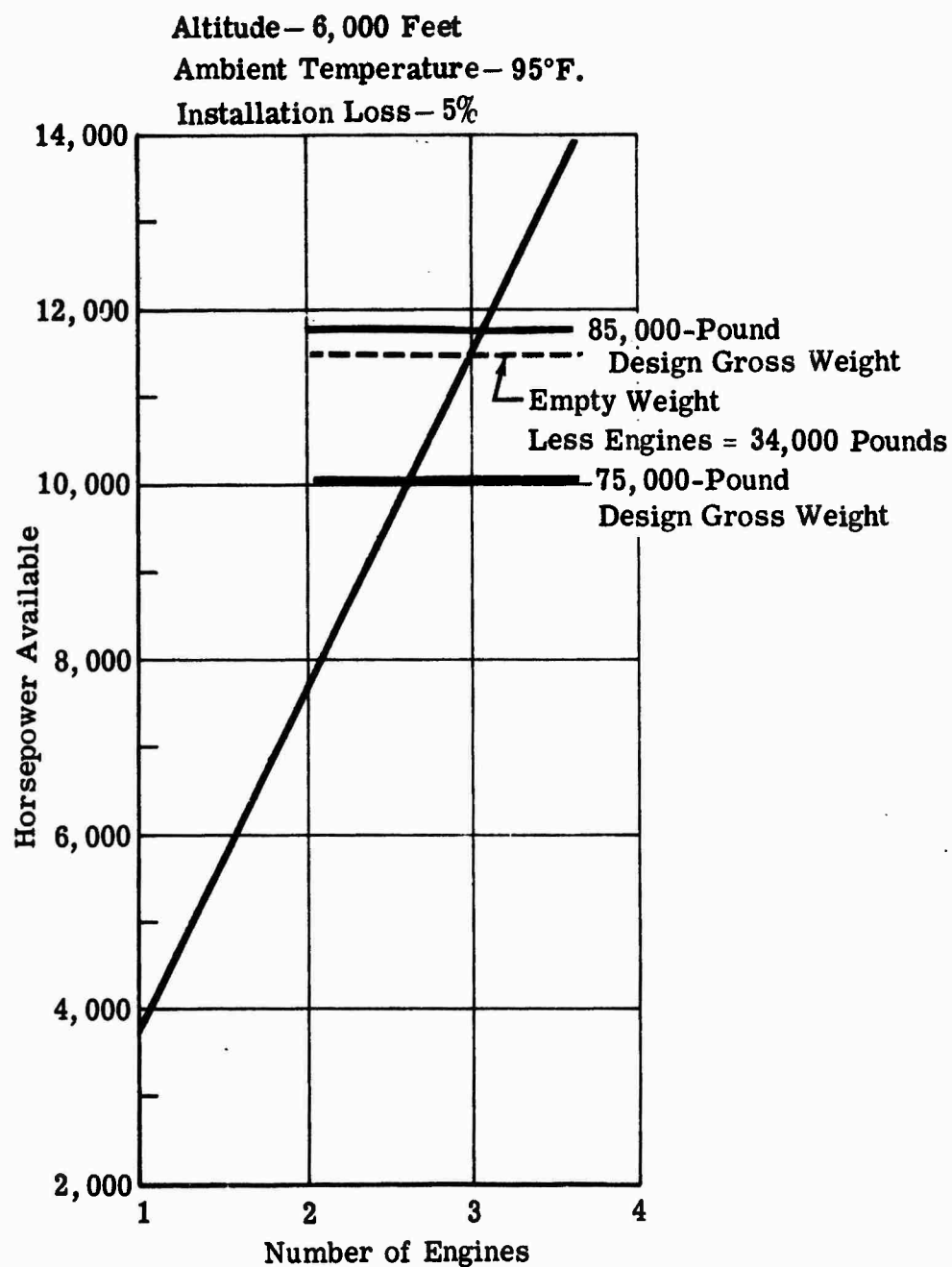
Figure 32. (U) Number of Engines Required— Model 501-M25
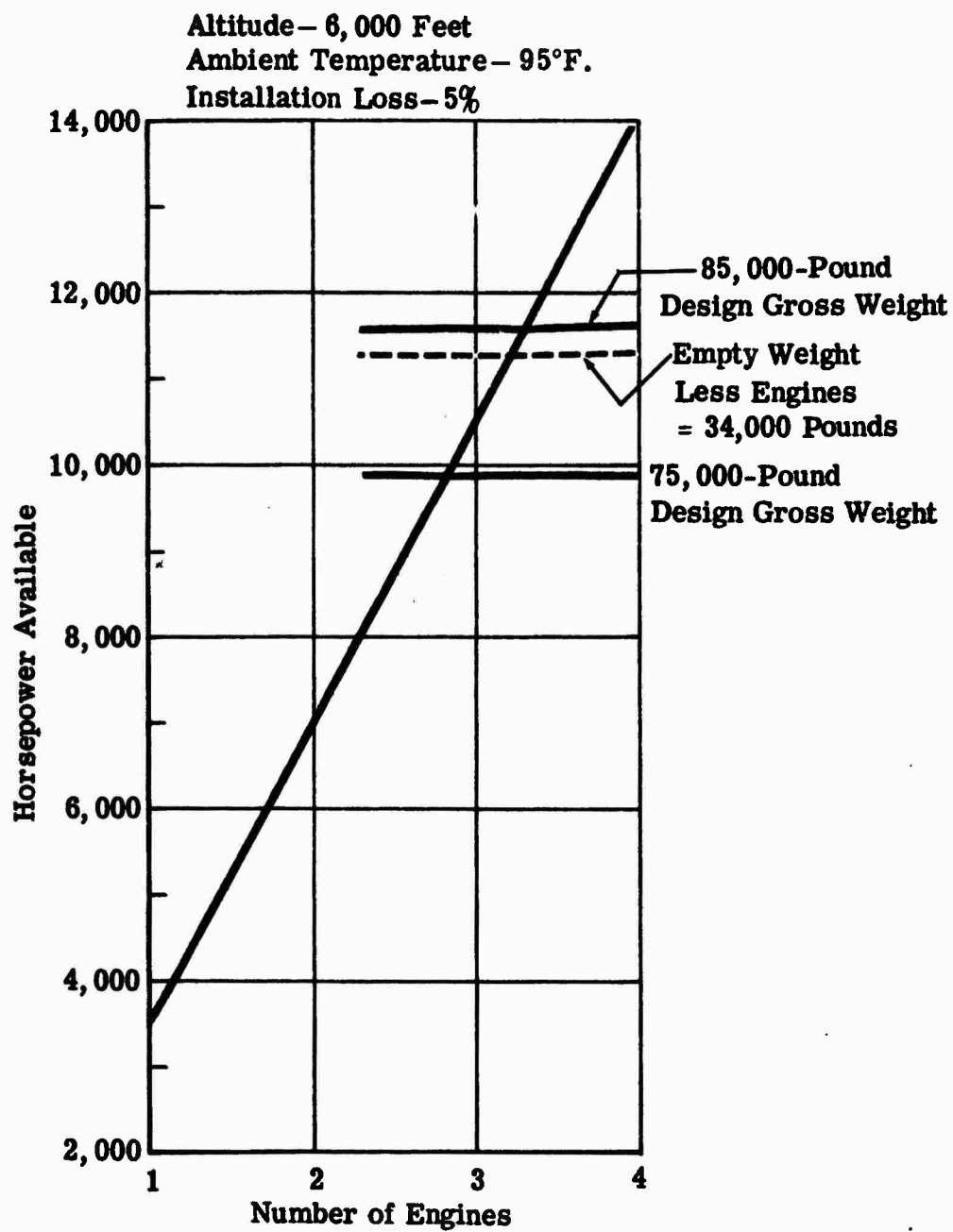
Figure 33. (U) Number of Engines Required—Model 501-M26

50

(U) The Model T78 engine which is now being developed has a 20 percent growth potential. The derivatives of this engine—Models 545, 546, and 548—are expected to have the same growth potential. This growth can be achieved without major redesign or development by increasing the turbine inlet temperature and airflow. Therefore, the power anticipated from the growth versions is also shown in Figures 29, 30 and 31.

(C) Figure 29 indicates that four Model 545 regenerative engines would be capable of supplying the power required for altitude hot-day hover for a helicopter with a heavy-lift design gross weight (DGW) of approximately 82,000 pounds. The growth version of this engine would provide power in excess of that required for a helicopter with a DGW of 85,000 pounds.

(C) Four Model 546 or 548 engines would be capable of supplying the power required for a helicopter with a DGW of 85,000 pounds. Use of three engines would require the use of approximately 30-percent power augmentation for transport altitude hot day hover.

(U) Figure 32 indicates that three Model 501-M25 engines could supply sufficient power for an HLH with an empty weight/DGW ratio of 0.47; however, a small degree of power augmentation would be required for an 85,000-pound helicopter.

(U) Use of three Model 501-M26 engines would require 11-percent power augmentation to meet the requirements of an 85,000-pound HLH. As indicated in Figure 33, no power augmentation would be required for a 78,000-pound helicopter.

(U) The results of this study indicate that a powerplant with a sea level standard-day static rating of approximately 18,000 shaft horsepower would be required for an 85,000-pound HLH. This value takes into consideration a 5-percent installation loss. Therefore, four engines rated at 4500 shaft horsepower or three engines rated at 6000 shaft horsepower would be required unless some means of power augmentation is employed.

(C) Gross Weight and Fuel Consumption (U)

(U) The gross weight and fuel consumption for the specified missions using various engine models are presented in Figures 34 through 40. These data are for an HLH with an empty weight (less engine weight) of

34,700 pounds, as explained in the Method of Analysis subsection. All data are based on sea level standard-day conditions.

(U) As shown in Figure 34, the gross weight for the heavy-lift mission varied from 82,600 to 84,900 pounds. This relatively small variation in gross weight indicates that engine and fuel weight are not predominant factors in this mission. This is because of the short range (20-nautical-mile radius) and the heavy payload (20 tons).

(U) The variation in gross weight for the transport mission—71,000 to 73,700 pounds—increased relative to that of the heavy-lift mission. However, the duration of the mission was not sufficient for specific fuel consumption differences to show a significant effect.

(U) As the helicopter range is increased and the fuel weight becomes a sizable percentage of the gross weight, the effect of specific fuel consumption becomes significant. As shown in Figure 36, the gross weight for the ferry mission ranged from 83,000 to 125,000 pounds. The superior specific fuel consumption of the regenerative engines, especially at partial power, showed a gross-weight reduction of 12,500 pounds when compared to the best nonregenerative configuration.

(C) One factor which must be considered in the overall analysis of the HLH is that of fuel consumption and its effect on logistics. The use of regenerative engines will result in fuel savings on all missions, as indicated in Figures 36 through 40. Table 6 shows the fuel savings resulting from the use of the Model 545-C2. Similar savings would result with a regenerative free-turbine engine.

TABLE 6
(C) FUEL SAVINGS WITH REGENERATION (U)

Mission	Fuel Savings (Pounds)
Heavy Lift	570
Special Heavy Lift	2450
Transport	1760
Ferry	21,000
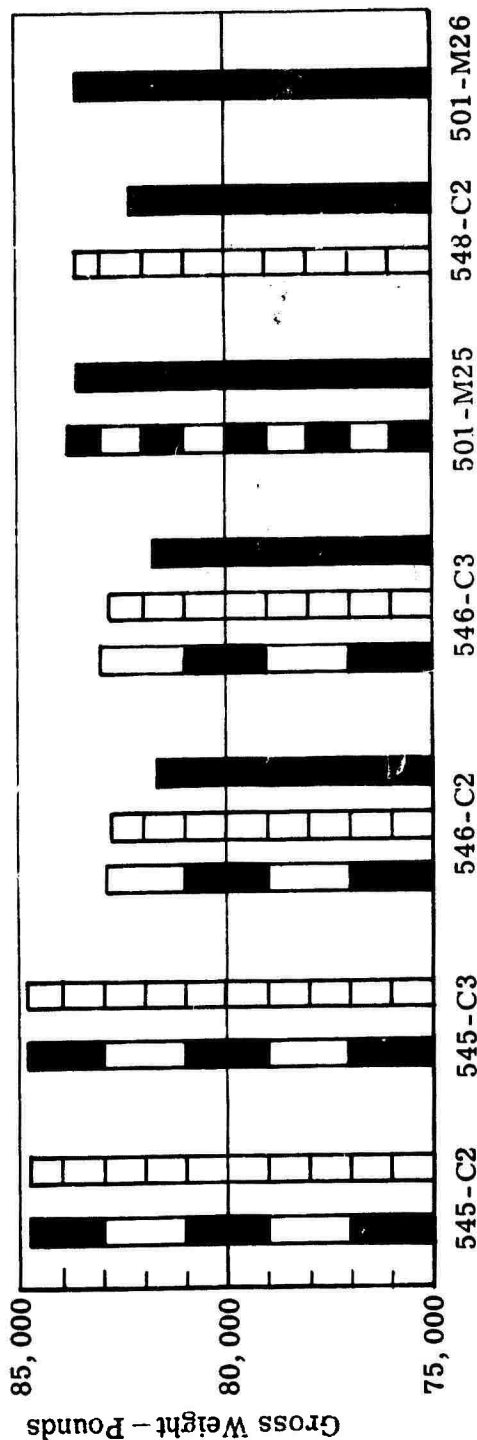
Figure 34. (C) Heavy-Lift Mission—Gross Weight (U)
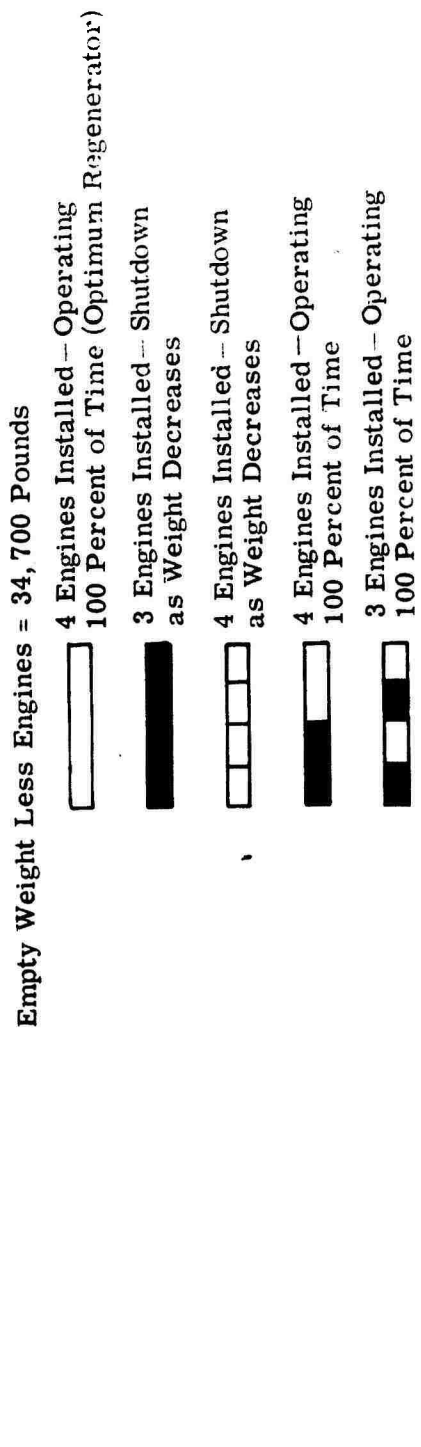
Figure 35. (C) Transport Mission—Gross Weight (U)

Empty Weight Less Engines = 34,700 Pounds

4 Engines Installed — Operating
100 Percent of Time (Optimum Regenerator)

3 Engines Installed — Shutdown
as Weight Decreases

4 Engines Installed — Shutdown
as Weight Decreases

4 Engines Installed — Operating
100 Percent of Time

3 Engines Installed - Operating
100 Percent of Time
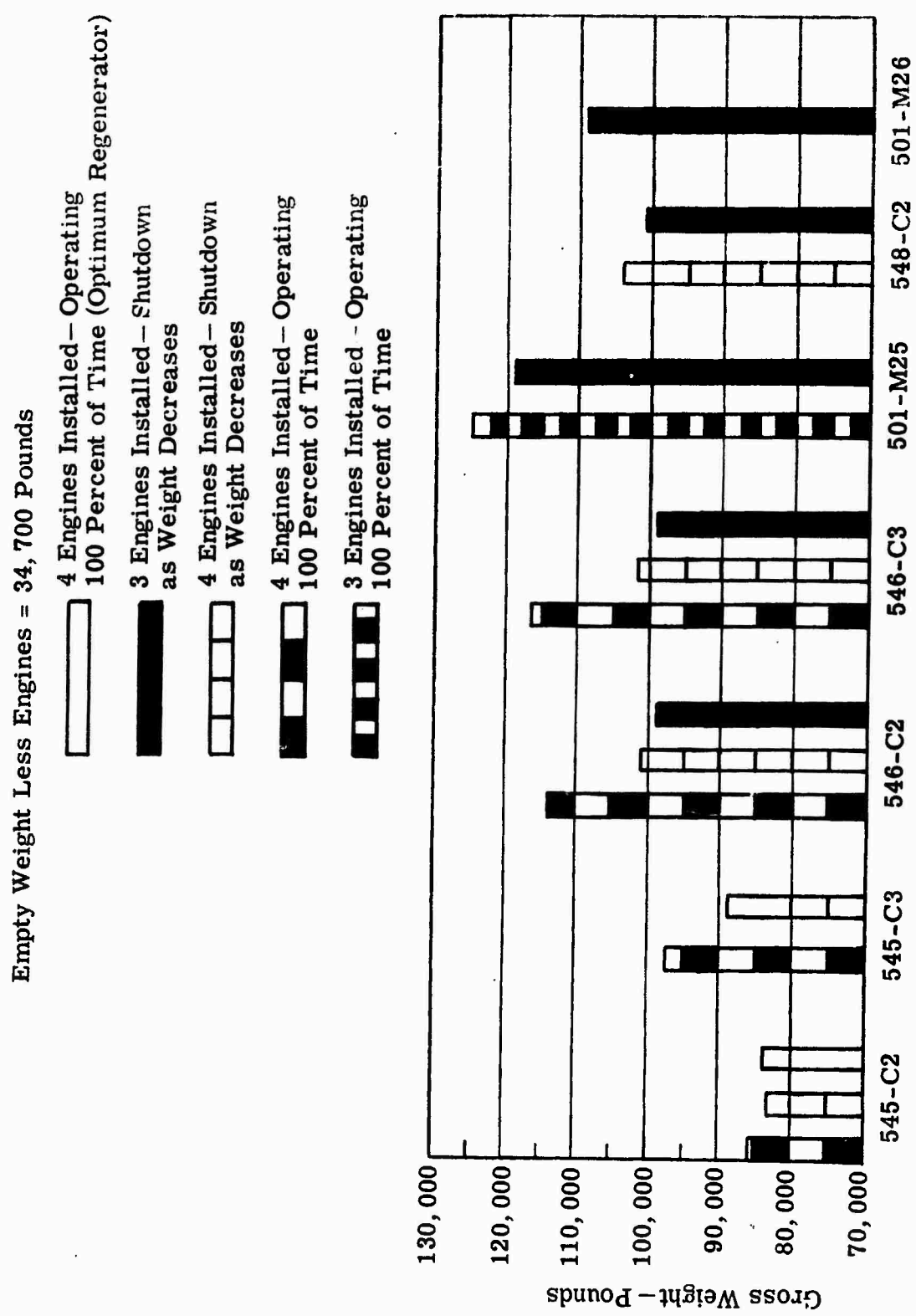
Figure 36.   (C) Ferry Mission—Gross Weight (U)
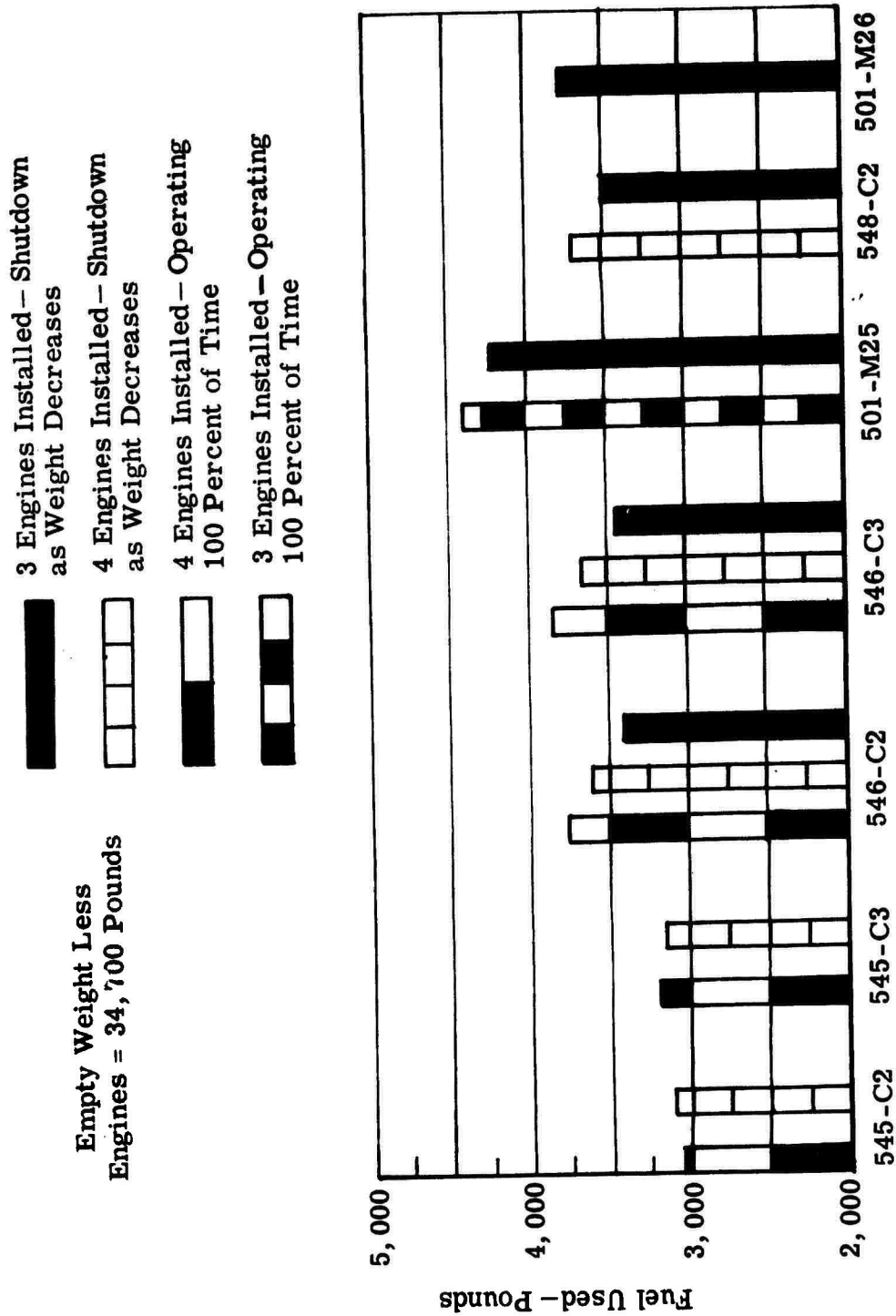
Figure 37. (C) Heavy-Lift Mission—Fuel Required (U)
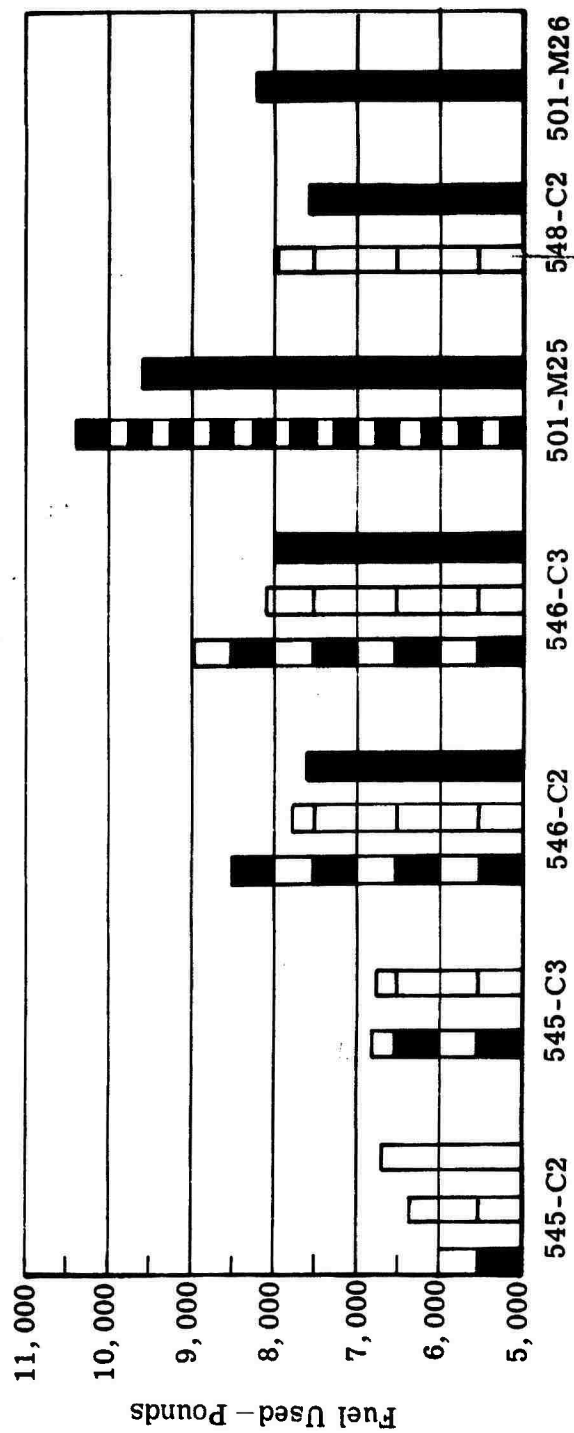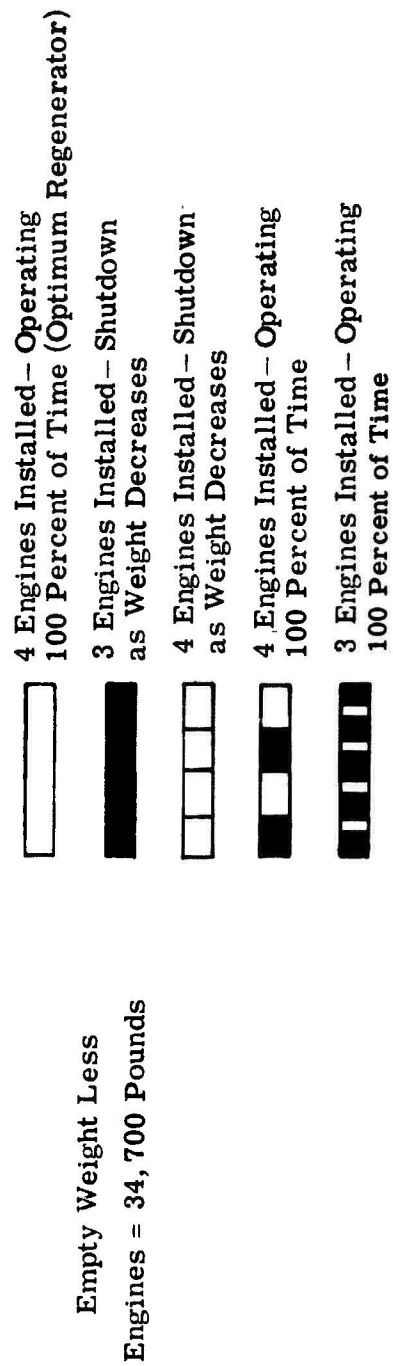
Figure 38. (C) Transport Mission—Fuel Required (U)

Empty Weight Less Engines = 34,700 Pounds

4 Engines Installed – Operating
100 Percent of Time (Optimum Regenerator)

3 Engines Installed – Shutdown
as Weight Decreases

4 Engines Installed – Shutdown
as Weight Decreases

4 Engines Installed – Operating
100 Percent of Time

3 Engines Installed – Operating
100 Percent of Time
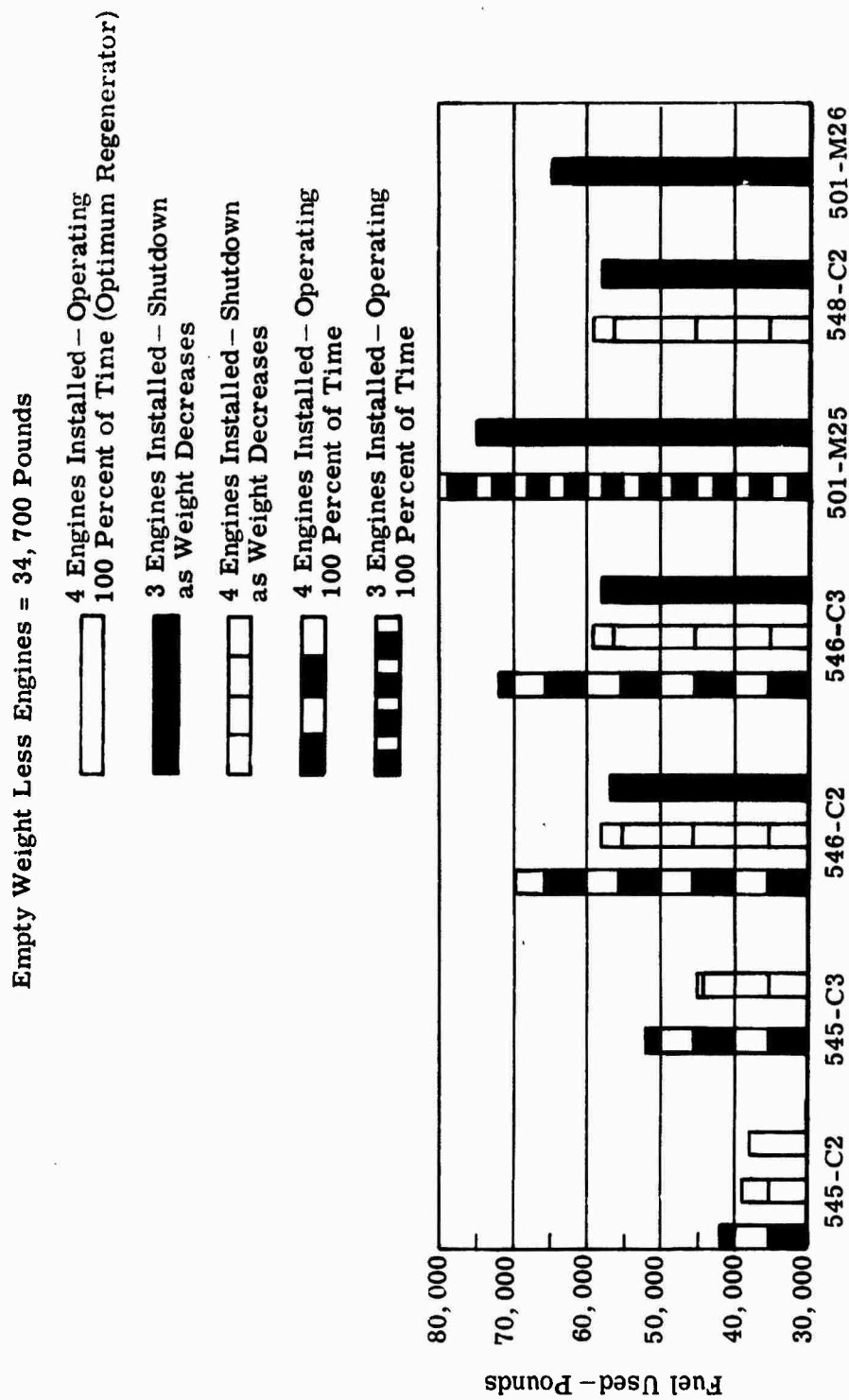
Figure 39.   (C) Ferry Mission—Fuel Required (U)

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Note: Engines are Shut Down as Gross Weight
Decreases Until Two Remain Operating.
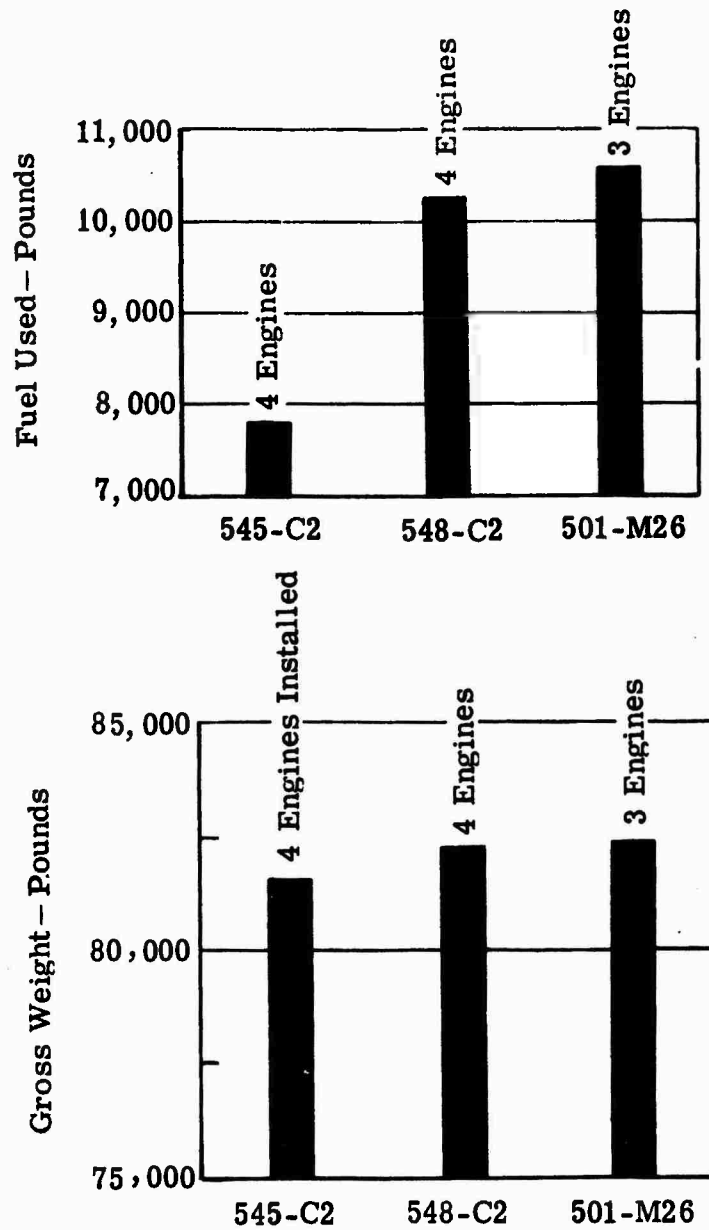
Figure 40. (C) Special 16-Ton, 100-Nautical-Mile Radius Mission (U)

(U) The effect of deliberate in-flight engine shutdown on range was investigated. This procedure has been successfully used with multi-engine, turbine-powered, fixed-wing aircraft to extend range. Use of this procedure requires that the remaining engines be operated at a higher power level, which usually results in a lower specific fuel consumption. In-flight shutdown was considered acceptable when the helicopter was capable of climbing at a rate of 100 feet per minute if one of the remaining engines ceased to operate. It was found that the use of this procedure would save fuel on all missions with nonregenerative engines. As much as 500 pounds, 1000 pounds, and 13,000 pounds of fuel, and therefore gross weight, could be saved on the heavy-lift, transport, and ferry missions, respectively.

(U) A reduction in the number of engines may offer an advantage from a maintenance, weight, and initial-cost viewpoint. Therefore, mission studies were conducted for three-engine helicopters, although there was insufficient installed power for the transport mission altitude hot-day hover. To meet this requirement, power augmentation would be necessary. Table 7 shows the savings in gross weight and fuel consumption when three rather than four Model 548-C2 engines are used.

TABLE 7
(C) EFFECT OF NUMBER OF ENGINES ON GROSS WEIGHT AND
FUEL CONSUMPTION—MODEL 548-C2 (U)

	Three Engines	Four Engines	Difference	Percent Change
Gross Weight—pounds				
Heavy Lift Mission	82,300	83,700	1,400	1.7
Transport Mission	70,400	72,000	1,800	2.5
Ferry Mission	100,900	103,700	2,800	2.7
Fuel Used—pounds				
Heavy Lift Mission	3,515	3,705	190	5.1
Transport Mission	7,540	7,965	425	5.4
Ferry Mission	58,200	59,830	1,630	2.9

(U) In-flight engine shutdown was used for both three- and four-engine studies. Therefore, the savings in fuel with the three-engine configuration was not as pronounced as would otherwise be expected.

(U) The data presented in Table 7 are for sea level standard-day missions. As shown in the subsection titled Power Augmentation, 880 pounds must be added to provide water-alcohol power augmentation for the transport mission altitude hot-day hover.

(C) To determine the effect of regeneration on ferry mission range, the lift-off weight of a helicopter utilizing Model 545-C2 (regenerative) engines was increased to that of a helicopter using Model 548-C2 (non-regenerative) engines. It was assumed that the difference in weight was fuel. Using this procedure, it was found that the 1500-nautical-mile ferry range was extended to 1950 nautical miles.

(C) The HLH design characteristic description specifies a design load factor of 2.5 for the heavy-lift mission. It also specifies that this factor could be reduced to 2.0 for the ferry mission. Therefore, the ferry mission lift-off weights of the Model 548 and Model 545 powered helicopters were increased to 125 percent of the heavy-lift-mission design gross weight. This increase in weight was made up of fuel. Use of this procedure increased the helicopter ferry range to 1514 nautical miles for the Model 548-powered helicopter and to 2012 nautical miles for the Model 545-powered helicopter, thus showing the marked ferry range increase associated with the use of regenerative engines.

(U) The preceding data were computed using an empty weight (less engines) of 34,700 pounds. This weight plus the installed weight of four 548-C2 engines will result in a heavy-lift mission empty weight/design gross weight ratio of 0.47. Figures 41 through 43 show the effect of empty weight/design gross weight ratio for the heavy-lift, transport, and ferry missions. By assuming an empty weight/gross weight ratio, it is possible to iterate for a gross weight and for fuel required—given any empty weight. Therefore, these curves may be used to extend the mission analysis with four Model 548-C2 engines.
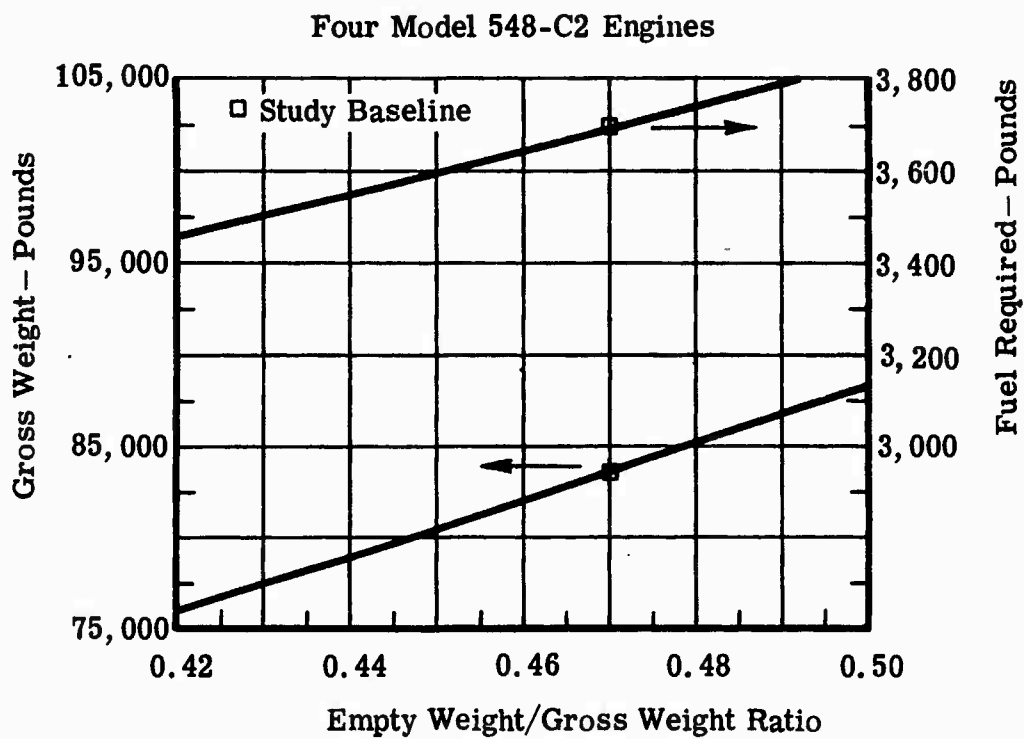
Figure 41. (C) Effect of Empty Weight/Gross Weight Ratio—
Heavy-Lift Mission (U)
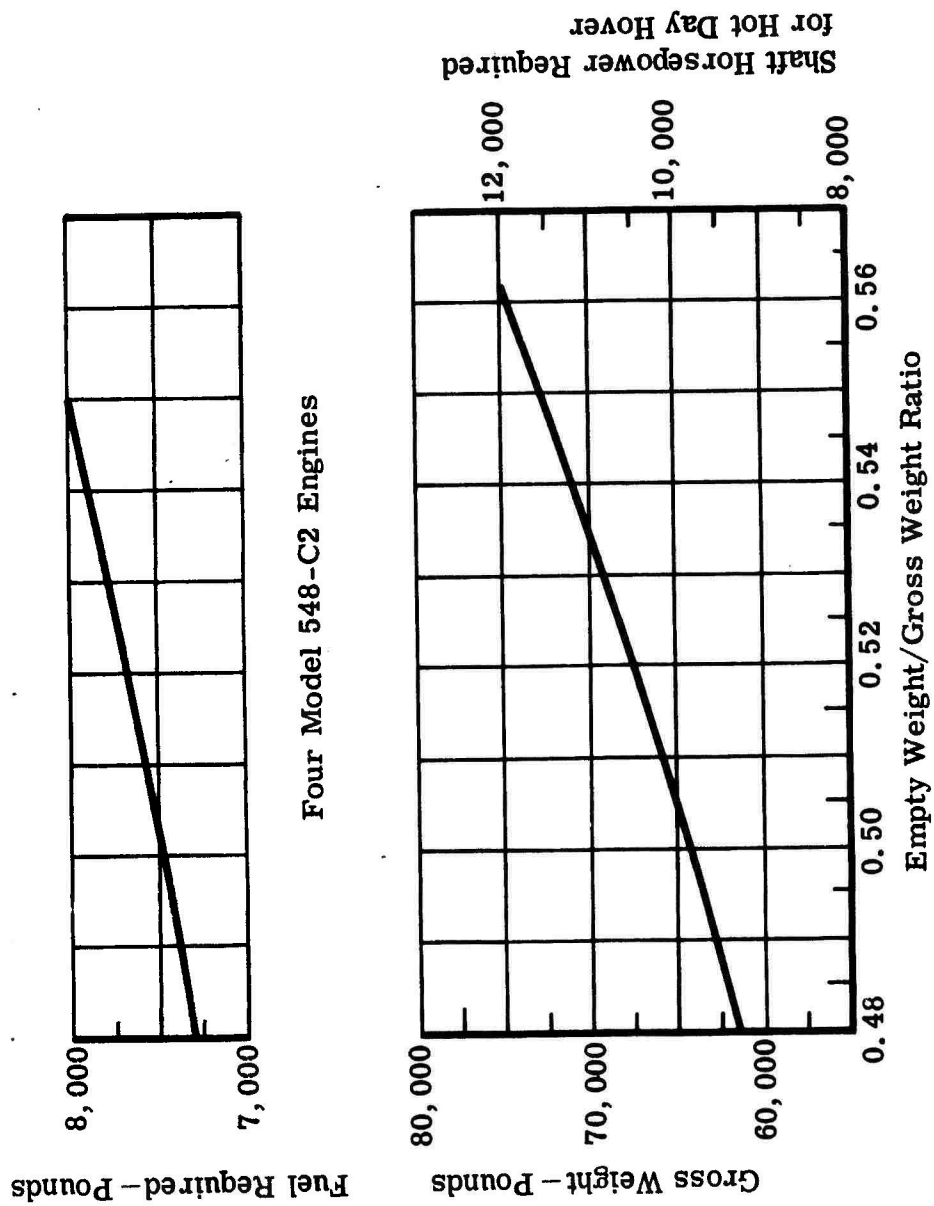
Figure 42. (C) Effect of Empty Weight/Gross Weight Ratio – Transport Mission (U)

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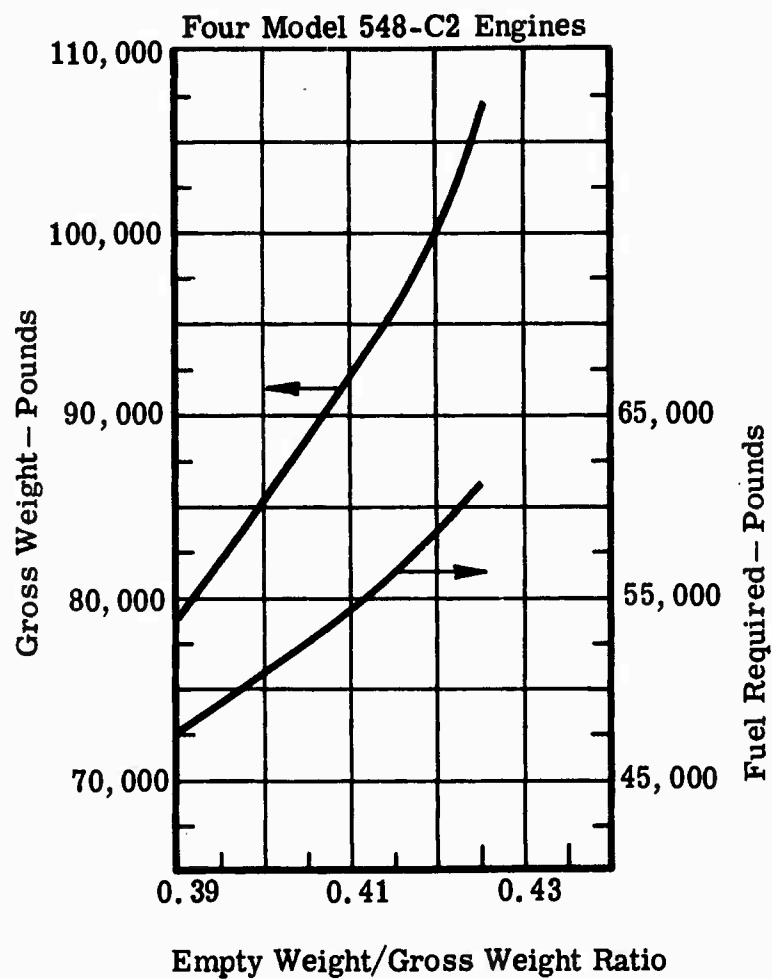
Figure 43. (C) Effect of Empty Weight/Gross Weight
Ratio—Ferry Mission (U)

(U) CONTROL-SYSTEM STUDIES

The control system studies were specifically directed toward the features of the HLH that were anticipated to have a significant influence on control system design and operation. These features included multiengine clustering with output shafts mechanically connected and a large helicopter rotor system with appropriate drive shafting. The purpose of these studies was to determine if these features imposed or produced any requirements or characteristics that might significantly influence control system, engine, or power transmission design. The studies included fixed-turbine, free-turbine, and regenerative engines. The areas of investigation are described in the following paragraphs.

CONTROL MODE

The multiengine configuration and the helicopter installation impose certain requirements that must be recognized by the control system. In the multiengine configurations considered, the output shafts of the engines are mechanically coupled in a common speed-reduction gearbox. The result is that during powered operation all engine output shafts operate at the same speed. Overrunning clutches can be employed between each engine power shaft and the helicopter rotor drive to allow automatic disconnect of an engine in the event its power level decreases to zero.

The HLH application dictates the following operational requirements that are pertinent to the engine control system:

- Automatic rotor speed governing
- Automatic control of engine power matching
- Rapid power transients
- Autorotation capability
- Ground idle operation with reduced rotor speed
- Ground idle operation with rotor not rotating
- Operation in flight with some engines shut down

The explanation of the control considerations involved in designing a system to meet these requirements is presented herein, including both free-turbine and fixed-turbine engines.

Rotor Governing and Control

The governing concept is straightforward. Engine power must be auto-matically regulated at a level to provide a specific output shaft speed. This can be accomplished by a control that directly senses the shaft speed and adjusts the engine power accordingly. The engine power can be changed by varying airflow and/or turbine inlet gas temperature. The load (power required) is established by the helicopter rotor which is directly regulated by the pilot. This section provides an explanation of the governing concepts relative to the free-turbine and fixed-turbine engines as associated with the helicopter application. Also included is an explanation of the steady-state and transient control automatic limit-ing functions required.

Free Turbine

The free-turbine engine allows varying the airflow along with the turbine temperature for power modulation at constant output shaft speed. The airflow variation is accomplished by varying gas producer speed. A turbine inlet temperature variation is required to achieve the variation in gas producer speed. The output shaft is connected to the power tur-bine. The free-turbine engine possesses certain steady-state and dy-namic characteristics that must be recognized in the selection of a governing mode. The criteria for the governing mode is that it must restrict steady-state rotor speed variations and maintain stable opera-tion throughout the power range. Also, it must effectively utilize the full engine potential during load transients to minimize transient rotor speed variations.

To conform with the requirements of the application, a power turbine governor must be employed to accomplish rotor speed control. The setting of this governor must be variable over a small range to provide rotor speed trim for optimum efficiency and/or coordination with col-lective pitch. A gas producer governor is required to provide limiting of the maximum gas producer speed during normal operation and to control the engine during ground idle where the helicopter rotor speed is not governed.

Figure 44 illustrates the steady-state control characteristics provided. With the gas producer governor at ground idle, the engine power will not be sufficient to reach rated rotor speed at minimum collective pitch (ground operation). With the gas producer governor set for limiting, the available power is sufficient to accelerate the rotor to a speed where

the power turbine governor is in control. The speed is governed by automatically reducing the available power (gas producer speed) to a value compatible with rotor load.

Because of the requirement for sensing two different spool speeds for control, the free-turbine control system must be divided into two separate components—the gas producer and power turbine governors. The gas producer control provides speed limiting, ground idle governing, and transient fuel limiting. This component in most systems contains the sole fuel metering valve in the system. The power turbine governor develops a governing signal to be transmitted to the gas producer control to affect fuel metering. The stability and response is then associated with the manner in which the power turbine governor signal is employed to regulate shaft horsepower. Control system gains and dynamics, in conjunction with the engine rotor gains and inertias, must be designed to meet the steady-state and dynamic requirements.

On many high-performance free-turbine engines, variable compressor vanes are employed to allow operation at high pressure ratio at maximum power while avoiding compressor surge at reduced power. In general, variable geometry can be controlled as a function of the gas
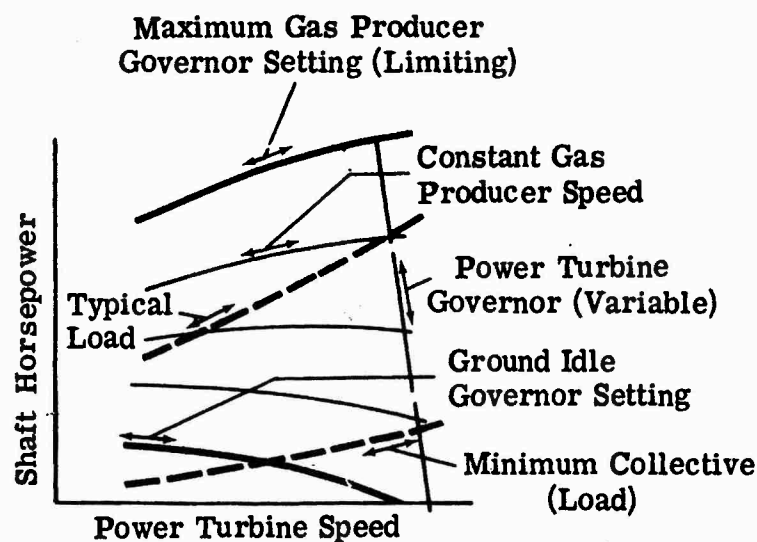
Figure 44. Free-Turbine Steady-State Characteristics

producer speed. However, in some proposed configurations, airflow variation is accomplished by varying the compressor geometry while maintaining constant gas producer speed (Allison Model 548-C2). In this case, the power turbine governor signal is employed to accomplish the variable geometry reset required as it varies metered fuel flow.

Fixed Turbine

The fixed-turbine engine is a single-spool engine that is mechanically connected to the helicopter rotor system. The engine must be operated at constant speed to provide the helicopter characteristics desired. The power level is changed through variation of the turbine inlet gas temperature with airflow remaining constant (e.g., Model T56).

Unlike the free-turbine engine, an engine speed change is not required to change power. Power transients require only a change in turbine gas temperature (fuel flow). There is basically only one acceptable way to regulate speed, and that is with a governor affecting the metered fuel flow. The power modulation required for governing of the fixed-turbine powerplant is achieved primarily by a governor operating on engine metered fuel flow. As rotor speed decreases below the governor setting, fuel flow is increased along with the available power. Similarly, a rotor speed increase will reduce the fuel flow and power, as shown in Figure 45.

There are certain engine designs that allow airflow variation while at constant engine speed and turbine inlet temperature. Airflow can be altered by variable compressor vanes providing power modulation (e.g., Model T78 turboshaft engine). In this case, the governor will be required to control both the fuel flow and compressor geometry.

The criteria for the governing mode are the same as for the free-turbine engine. The controls must maintain a minimum steady-state rotor speed variation and stable operation throughout the power range while providing sufficient power response to minimize transient rotor speed variations. Since an engine speed variation is not required for a power change, the transient power response is dependent only on the response of the fuel and variable compressor geometry control systems. A governor is required that directly senses the engine (and rotor) speed and automatically regulates the variable compressor vanes and/or fuel flow accordingly. The governor speed setting must be resettable over a small speed range, similar to the free-turbine mode.
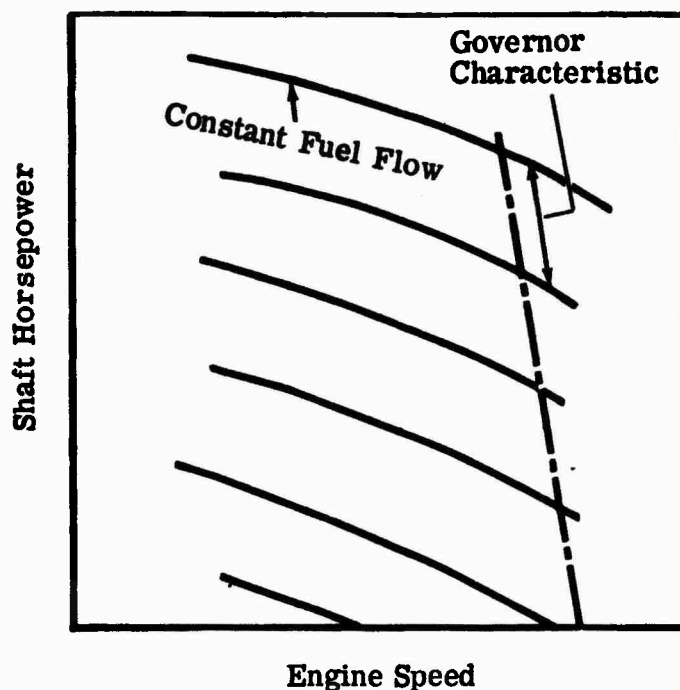
Figure 45. Fixed-Turbine Steady-State Characteristics

Automatic Limiting

To protect the engines from exceeding their structural limitations, the
control system must automatically limit their operation. A turbine
temperature limiter could be employed to prevent steady-state opera-
tion at temperatures greater than the level established for structural
integrity and long life. Closed-loop turbine temperature limiting is
accomplished by sensing the gas temperature with thermocouples, us-
ing an electronic amplifier control which compares the sensed tem-
perature signal to a known reference signal establishing the turbine
temperature error. This error signal is then transmitted to the fuel
control to limit the fuel flow accordingly. This feature is applicable to
the single governor and the individual governor concepts, both for free
and fixed-turbine engines. The free-turbine engine control would pro-
vide a gas producer governor to limit its speed. This would be accom-
plished by regulating the fuel flow to prevent exceeding the maximum
gas producer speed.

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During engine power transients, engine operation must be limited to prevent compressor surge, limit the turbine temperature, and prevent burner blowout. This is conventionally accomplished through open-loop scheduling of the fuel flow and variable compressor geometry as a function of the engine speed and air density. The HLH application does not alter these requirements from those of other applications (such as fixed wing).

Regenerative Engines

The governing requirements and concepts are not affected by the employment of regeneration on the fixed turbine and free turbine engines. The air density compensation must be different for the regenerative engines to compensate for dynamics and variations in the regenerator. The scheduling parameters and type of scheduling required are very similar to those employed on the Model T78 engine and are discussed further in the subsection titled Fuel Control Operation During Starting.

Clustered Multiengine Governing

In the clustered multiengine power system for the HLH, the output shafts are mechanically geared together so that during normal operation they are at identical speeds. This is true whether they are free turbine or fixed turbine engines. There are two approaches to rotor speed governing that are applicable to a mechanically coupled multiengine power system. They are applicable to either the free turbine or fixed turbine and the nonregenerative or regenerative engines. In one concept, each engine is controlled by its own governor, thus requiring individual governors. In the other concept, a single governor, mounted on the combining gearbox, is employed to reset all engine controls equally and simultaneously. The reasons for considering the single governor approach is that it would simplify the system, improve the governing accuracy so that closed-loop load sharing would not be required, be suitable to isochronous governing, and provide an improvement in governing stability. The purpose of this subsection, therefore, is to present an evaluation of the single and individual governor concepts.

Governor Operation

In all cases, the control systems must be designed so that they obtain maximum power from all engines when required. Also, zero horsepower operation of all engines must be obtainable when required.

Figure 46 illustrates the control concepts for free turbine engines.
For both the single and individual governor approaches, separate gas
producer controls and electronic temperature controls are required for
each power section. This is necessary since the different engines in
the system will not be identical in their steady-state or transient char-
acteristics. The control and limiting of each engine, therefore, must
be based on parameters sensed specifically at each individual gas pro-
ducer. The throttle lever input is required to operate the mechanical
fuel cutoff and to establish the gas producer governor setting for limit-
ing and for ground idle operation. The gas producer speed is sensed to
enable proper limiting during accelerations and to enable governing at
ground idle and limiting of the maximum speed. Compressor discharge
pressure and compressor inlet temperature are sensed to provide an
air density bias on the transient fuel schedules. A compressor dis-
charge pressure sense is preferred since it provides for corrective
action in the event of a compressor surge and is not subject to probe
icing which can occur at the compressor inlet. The temperature can
be sensed at either the compressor inlet or discharge. In theory the
discharge temperature allows more accurate scheduling; however, it
requires sensing the gas of a moderately high temperature level and
large temperature range. Also, discharge temperature sensing can
impose undesirable lags during engine speed transients due to sensor
and servo system lag. Compressor inlet temperature sensing is de-
sired because it offers a more favorable temperature level and range
in addition to an insensitivity to engine speed.

Turbine temperature limiting is accomplished by a signal generated in
the electronic temperature controls and transmitted to the gas producer
controls. This limiting is effectively accomplished by reducing the fuel
flow on the specific engine involved to alleviate the overtemperature
condition.

The input (reset) to the gas producer control is developed in the power
turbine governor to provide fuel regulation during rotor speed govern-
ing. The only difference between the single and the individual governor
approaches is in the origin of the reset signal. In the single governor
approach, the governor is mounted on the combining gearbox (between
the overrunning clutches and the rotor), and its signal is simultaneously
and equally transmitted to all of the gas producer controls. As rotor
(power turbine) speed varies, a reset signal, as shown in Figure 47, is
generated by the governor. A limit on the governor reset is provided
to prevent excessively low turbine temperatures and gas producer
speed during decoupled, high rotor speed operation.

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In the individual governor system, three separate power turbine governors are employed (on a three-engine power system), with each independently generating a reset signal for transmission to a specific gas producer control. The power turbine governors are mounted on accessory gear drive systems of the power turbine.

In the fixed-turbine system, there is no specific requirement that the rotor governor be a component separate of the fuel control. In fact, on a single-engine application, this results in unnecessary complexity. Figure 48 illustrates the single- and individual-governor concepts for a fixed-turbine engine. The conventional approach is to employ individual governors that are contained in the fuel controls. When considering a clustered engine application, however, there are certain advantages of a single governor (relative to load sharing and isochronous operation) that require further evaluation. Similar to the free-turbine engine, a single governor is mounted on the combining gearbox. The output signal generator is mounted on the combining gearbox. The output signal is transmitted to the individual fuel controls to regulate their fuel metering valves. Figure 49 illustrates the type of scheduling that could be employed in a single-governor approach. Engine speed must also be sensed in each fuel control to provide proper scheduling during starting and engine acceleration. This is necessary because the system must be capable of individual engine startup while decoupled from the combining gearbox.

Compressor air pressure and temperature signals are employed to bias the fuel scheduling according to air density. Compressor discharge pressure (CDP) and temperature (CDT) on a nonregenerative engine theoretically provide the most accurate scheduling of fuel flow to achieve a turbine temperature. Use of these fuel scheduling parameters may be most desirable for HLH application. The turbine inlet temperature is sensed for use in providing closed-loop limiting of each individual power section.

The gas producer controls and the engines do not possess identical characteristics. The result of this, illustrated in Figure 50, is characteristic for single governor or individual governors, and for fixed- and free-turbine engines. At the high power levels and small reset signals, the engines are operating at high turbine temperatures. If engine 1 reaches the turbine temperature limiter setting and the power turbine speed further droops off (increased load), the power level of engine 1 will not increase. However, reducing the reset signal will

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Single Governor

T/L N_1 CIT   CDP

Steady State Control | Transient & Overspeed Limiting | Turbine Temperature Limiting

Gas Producer Control No. 1

Power Turbine Speed →

Governor Speed Setting →

Single Power Turbine Governor

Reset →

Gas Producer Control No. 2

Gas Producer Control No. 3
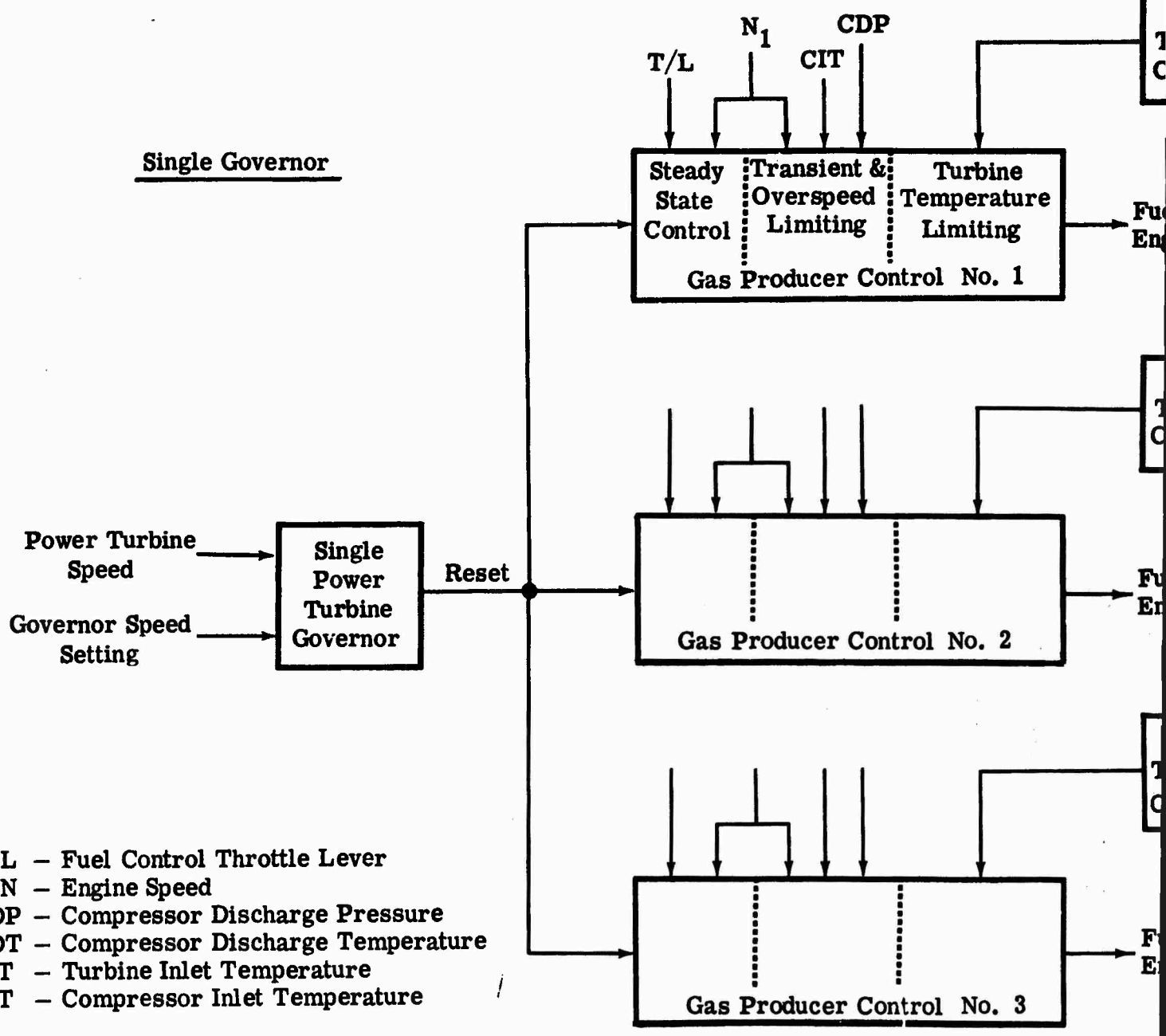
T/L — Fuel Control Throttle Lever
 N — Engine Speed
CDP — Compressor Discharge Pressure
CDT — Compressor Discharge Temperature
TIT — Turbine Inlet Temperature
CIT — Compressor Inlet Temperature

Figure 4

73

1

TIT

tronic
erature
ol No. 1

Individual Governor

Power Turbine
Speed

Governor Speed
Setting

ow To
No. 1

Power
Turbine
Governor
No. 1

T/L   N₁   CIT   CDP

| Steady State Control | Transient & Overspeed Limiting | Turbine Temperature Limiting |

Gas Producer Control   No. 1

T
C

F
E

tronic
erature
ol No. 2

ow To
No. 2

Power
Turbine
Governor
No. 2

Gas Producer Control No. 2

T
C

F
E

tronic
erature
ol No. 3

ow To
No. 3

Power
Turbine
Governor
No. 3

Gas Producer Control   No. 3

T
C

ree-Turbine Control Concepts

2

T/L N_1 CDP

CIT

TIT

Electronic
Temperature
Control No. 1

Power
Turbine
Governor
No. 1

Steady
State
Control

Transient &
Overspeed
Limiting

Turbine
Temperature
Limiting

Gas Producer Control No. 1

Fuel Flow To
Engine No. 1

Electronic
Temperature
Control No. 2

Power
Turbine
Governor
No. 2

Gas Producer Control No. 2

Fuel Flow To
Engine No. 2

Electronic
Temperature
Control No. 3

Power
Turbine
Governor
No. 3

Gas Producer Control No. 3

Fuel Flow To
Engine No. 3

3

Figure 47. Power-Turbine Governor Characteristic

continue to increase the power levels of the engines that are not temperature limiting. Thus, the maximum available power is the total that all three engines can develop when operating at maximum turbine temperature.

Similarly, as shown in Figure 51, all three engines will not reach zero power and decouple at the same reset signal level. Engines 2 and 3 can decouple while the rotor system is still requiring positive power from engine 1. A further increase in rotor speed is required to obtain decoupling of all engines from the rotor system.

The result of this difference between controls and engines is a higher steady state rotor speed variation to cover the range of powers from maximum to zero power. Also, load sharing by each engine is not equal. The gas producer controls and power turbine governor must be designed with sufficient range of reset to enable operation of the complete power system from zero to maximum power.

Governing Accuracy

A study of multiengine governing was conducted comparing a single high gain governor with three separate governors to simultaneously

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and equally reset three low-gain controls (on a three-engine configuration). The purposes of the study were (1) to determine which concept would provide the most accurate matching of engines when considering production control scheduling tolerances, and (2) to determine the magnitude of their differences to allow evaluation of the concepts. The analysis was conducted for both the free turbine and the fixed turbine engines and was based on a governor designed with a nominal droop of 5-percent speed from maximum to zero power.

Fixed Turbine

The fixed-turbine fuel flow must be varied from 100 percent at maximum power to approximately 30 percent (of maximum flow) at zero power and constant 100-percent engine speed. Experience has indicated that suitable control governors with a 5-percent total droop will have a gain variation in production equivalent to plus or minus 0.5-percent speed at minimum flow when set exactly on point at maximum flow. This will result in a maximum potential difference of 12.5 percent (of maximum fuel flow) between any two governors. In addition to governor gain variation, hysteresis also affects governor variations. Assuming a total hysteresis of 0.25-percent speed, the variation of fuel flow will be 3.5 percent (of maximum) throughout the speed range. At the minimum power condition, this is additive with the gain variation effect, producing a total potential engine-to-engine fuel flow variation of 16.0 percent (of maximum). See Figure 52.

By using a single, high gain governor to develop and transmit a signal to operate three separate fuel valves, the governor gain and hysteresis tolerance is eliminated in terms of its effect on matching of the output of the three control systems. The relevant control tolerance is that of scheduling fuel flow as a function of reset signal. A typical tolerance, including hysteresis, for this low gain scheduling mechanism is plus or minus 2 percent of maximum fuel flow over the entire flow range. Thus the variation in fuel flows between the engine controls could be 4 percent over the full power range. See Figure 53.

A comparison of the two governing concepts is presented in Table 8 for the fixed-turbine engine.

The variations in power indicated are solely due to the control variations and do not include engine variations. At maximum power, the two concepts are essentially of equal accuracy. However, at low powers the single governor is more accurate, allowing only one-fourth of the fuel flow variation of individual governors.

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Single Governor

Engine Speed →
Governor Speed Setting →
Single Governor

T/L — Fuel Control Throttle Level
N — Engine Speed
CDP — Compressor Discharge Pressure
CDT — Compressor Discharge Temperature
TIT — Turbine Inlet Temperature

Individual Governors
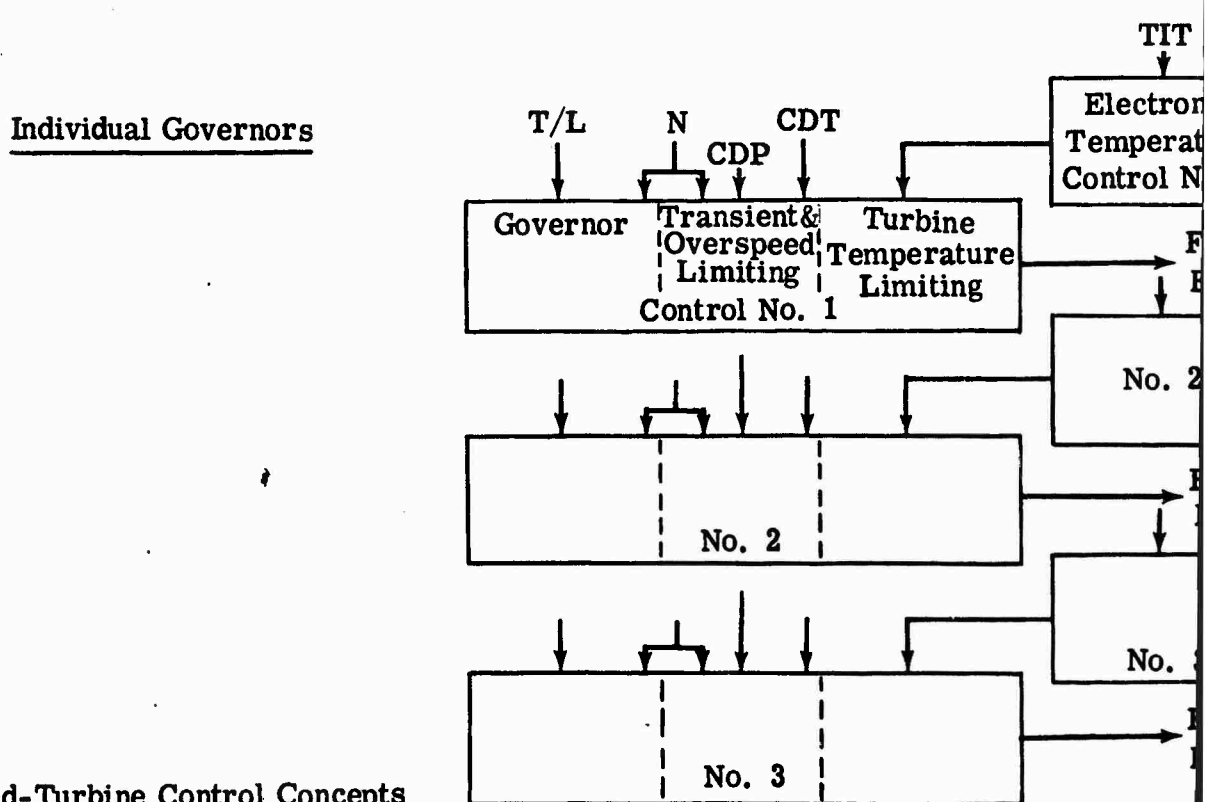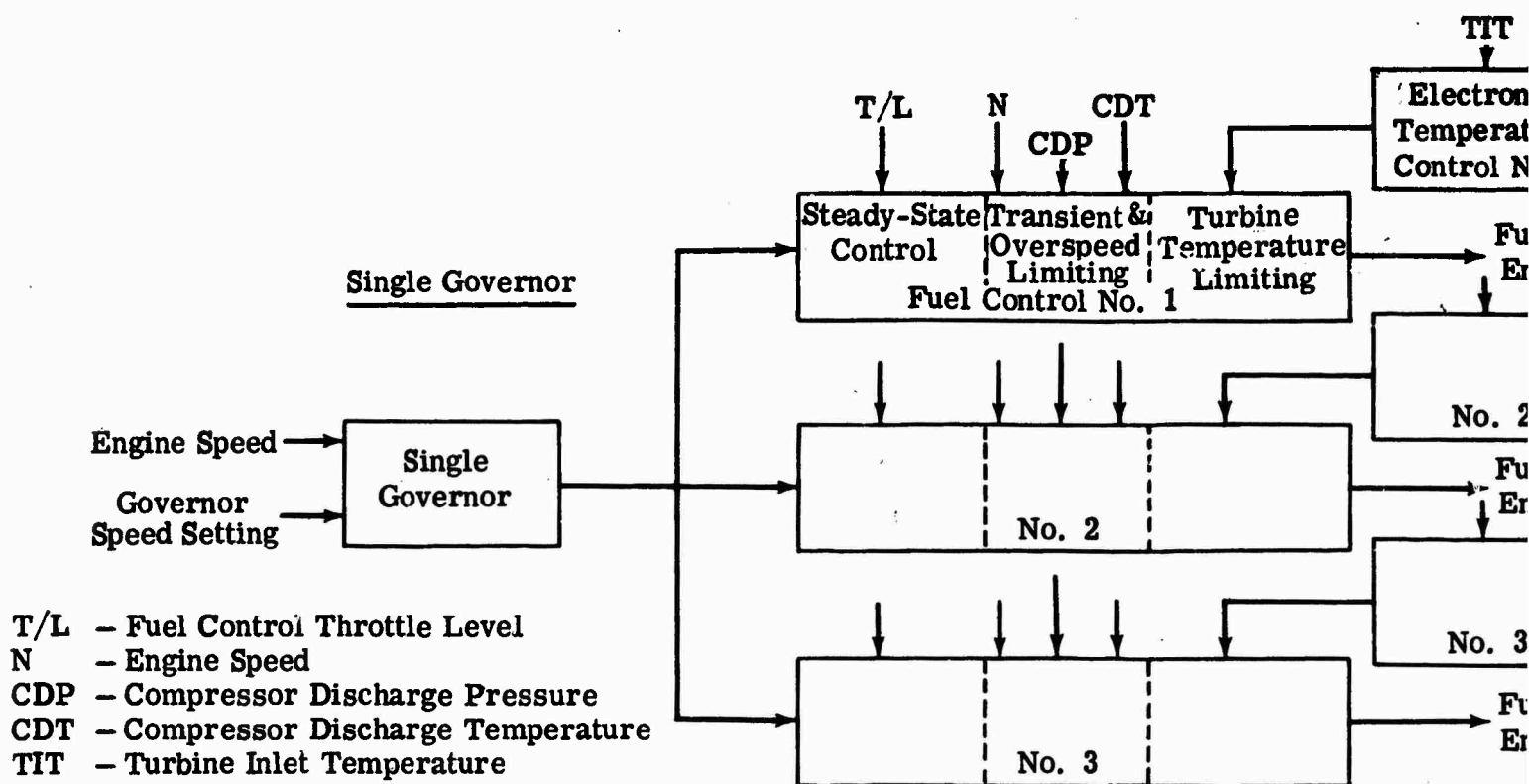
Figure 48.   Fixed-Turbine Control Concepts

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Single Governor

T/L N CDP CDT TIT

Electronic Temperature Control No. 1

Steady-State Control — Transient & Overspeed Limiting — Turbine Temperature Limiting

Fuel Control No. 1

Fuel Flow To Engine No. 1

No. 2

Single Governor

No. 2

Fuel Flow To Engine No. 2

No. 3

Throttle Level

Discharge Pressure
Discharge Temperature
Temperature

No. 3

Fuel Flow To Engine No. 3

Individual Governors

T/L N CDP CDT TIT

Electronic Temperature Control No. 1

Governor — Transient & Overspeed Limiting — Turbine Temperature Limiting

Control No. 1

Fuel Flow To Engine No. 1

No. 2

No. 2

Fuel Flow To Engine No. 2

No. 3

No. 3

Fuel Flow To Engine No. 3

xed-Turbine Control Concepts

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Figure 49.   Fixed-Turbine, Single Governor Scheduling
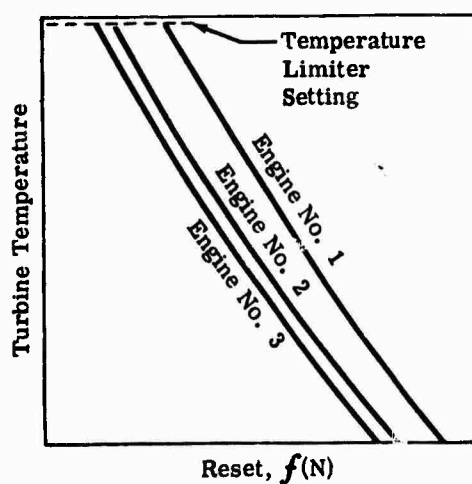
Figure 50.   Effect of Control and Engine Variations on
Turbine Temperature
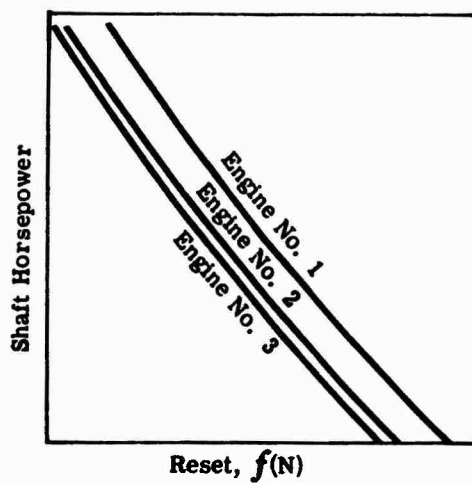
Figure 51. Effect of Control and Engine Variations on Shaft Horsepower
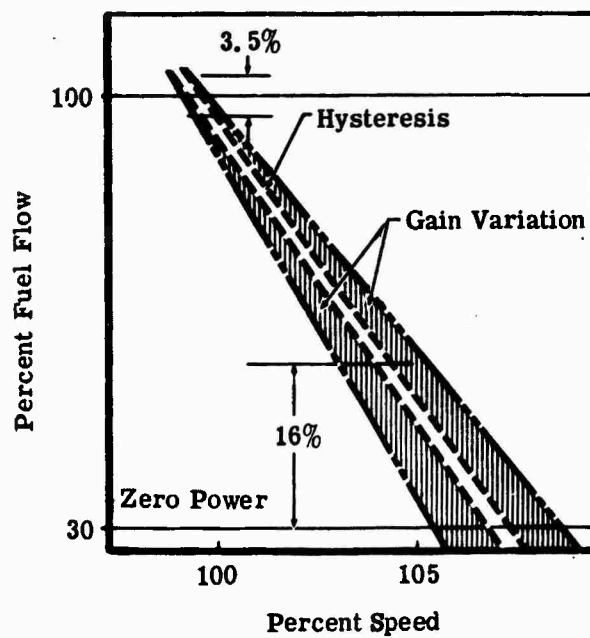
Figure 52. Control Variations With Individual Governors—Fixed Turbine
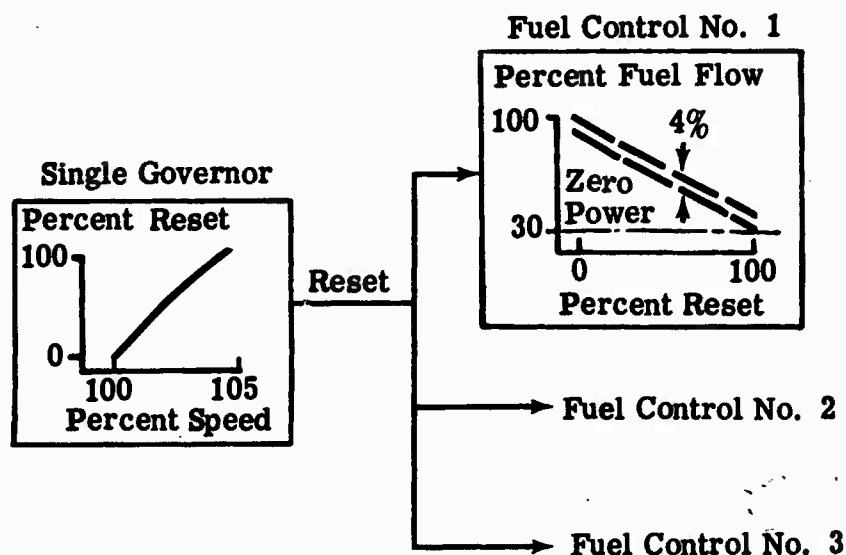
Figure 53. Control Variations With Single Governor—Fixed Turbine

TABLE 8
COMPARISON OF TOTAL FUEL FLOW AND TOTAL POWER
VARIATIONS—FIXED TURBINE

Governor Concept	Power Level	Total Fuel Flow Variation (% of Maximum)	Total Power Variation (% of Maximum)
Single Governor	Maximum	4.0	5.7
	Minimum	4.0	5.7
Individual Governors	Maximum	3.5	5.0
	Minimum	16.0	23.0

Free Turbine

The free-turbine control in the HLH will employ the reset mode whereby the power turbine governor develops a signal to reset the gas producer control. This governing mode is applicable to both the single-governor and individual-governor concepts.

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The individual-governor approach involves the summation of the tolerances of the power turbine governor (both gain variations and hysteresis) and the reset mechanisms of the gas producer control. Similar to the governor for the fixed turbine engine, the gain variation is equivalent to plus or minus 0.5-percent power turbine speed at zero power (maximum reset), assuming precise speed setting at maximum power (zero reset). Also, a governor hysteresis of 0.25-percent power turbine speed over the full reset range (Figure 54) would be encountered. The gas producer reset scheduling accuracy, including hysteresis, is in the order of 1 percent in speed over the complete range. For an engine that utilizes a gas producer speed variation of from 100 percent at maximum power to 80 percent at zero power (20-percent speed range), the maximum production variation could cause as much as a 6 percent gas producer speed difference between engines at low engine powers. At maximum power, the speed difference is 1.5 percent.

The single-power-turbine-governor approach eliminates the effect of production variations of this high gain device, thus greatly improving system accuracy. In this concept, a single governor mounted on the combining gearbox will equally and simultaneously reset all three gas producer controls. The variations in the reset mechanisms would be the sole contribution by the control system to unbalancing the engine powers. (See Figure 55.) This concept results in a difference between engines of approximately 1 percent in gas producer speed throughout the power range.

A comparison of the two governing concepts is presented in Table 9.

TABLE 9
COMPARISON OF TOTAL FUEL FLOW AND TOTAL POWER
VARIATIONS—FREE TURBINE

Governor Concept	Power Level	Total Gas Producer Speed Variation (% of Maximum)	Total Power Variation (% of Maximum)
Single Governor	Maximum	1.0	5.0
	Minimum	1.0	5.0
Individual Governors	Maximum	2.0	10.0
	Minimum	5.5	27.5
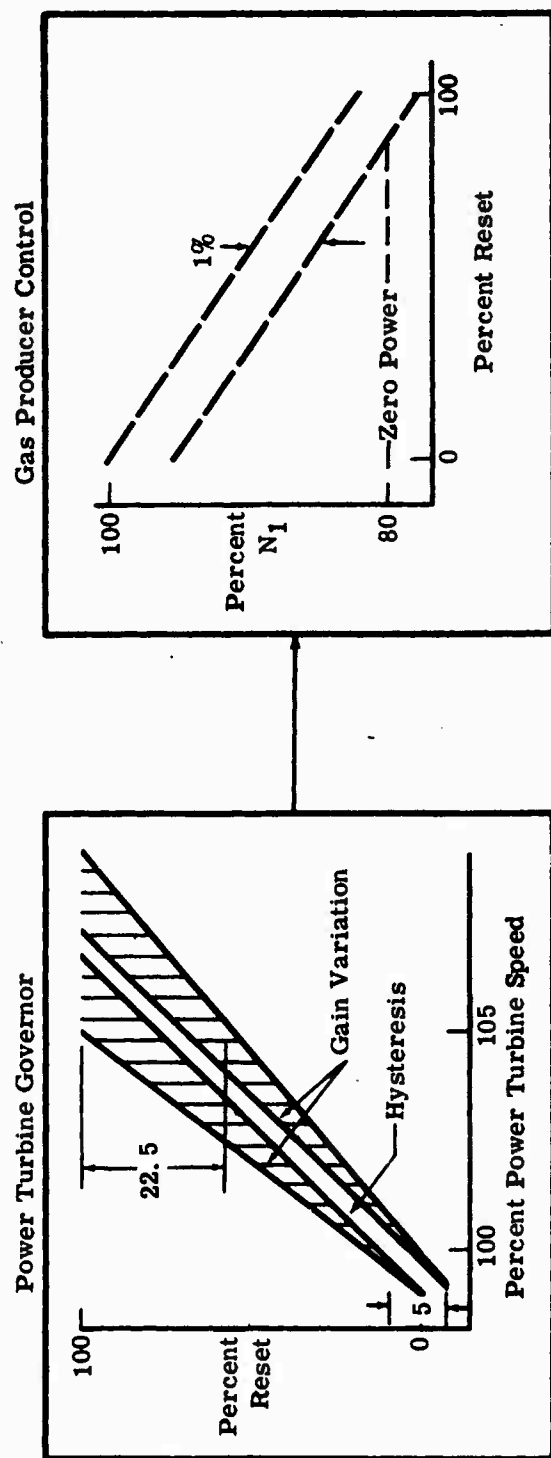
Figure 54. Control Variations With Individual Governors—Free Turbine

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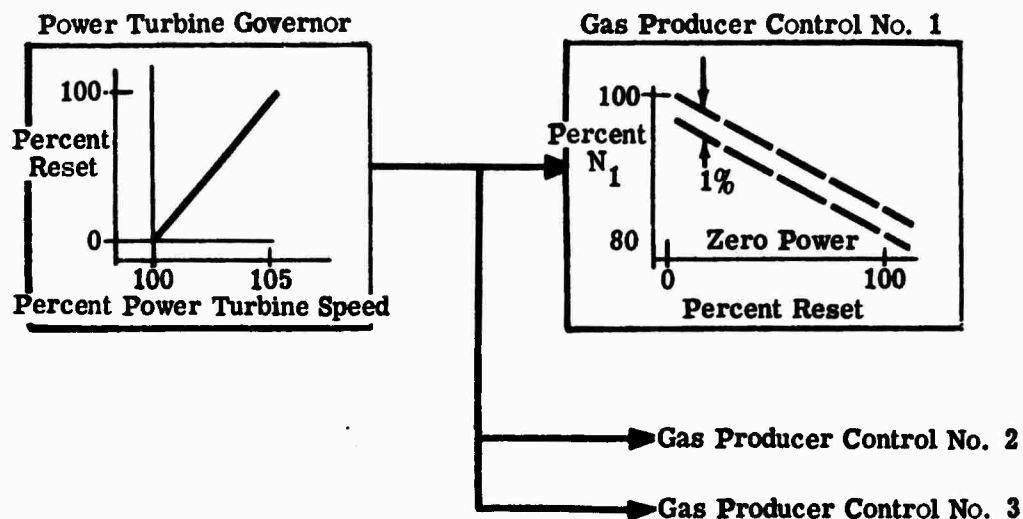
Figure 55. Control Vari tions With Single Governor—Free Turbine

Assuming a linear power-gas producer speed relationship at minimum
power the 5.5-percent gas producer speed would be equivalent to 27.5-
percent power variation between engines with individual power turbine
governors. Power variation with a single governor would be in the
order of 5 percent at minimum power, one-sixth of that of the in-
dividual-governor approach.

In summation, the single-governor approach would provide a significant
improvement in control system matching on a multiengine powerplant.
This is true for both the fixed- and free-turbine powerplants.

System Considerations

The purpose of this subsection is to introduce certain system features
that require consideration in governing. One of these is the effect of
engine layout on the governor system. With the free turbine engine
there are two configurations that appear prominent. In one the power
turbines are mounted on the aft end of the gas producer with their out-
put shafts connected to the combining gearbox. In the other, the power
turbines are mounted on the combining gearbox, with the gases from

each gas producer ducted and confined to their specific power turbine. In both configurations, the gas producer exhaust gases are ducted and confined to their specific power turbine.

With the latter engine design the governor(s) must be mounted on the combining gearbox because the turbines are an integral part of this assembly. The gearbox-mounted turbine design could utilize a somewhat simpler governing system by the use of a single governor mounted on the combining gearbox. The output of this governor would be transmitted (probably hydraulically) to all gas producer controls. A single governor can be employed if the speed sensing location is such that the overrunning clutches are between the power turbines and the speed pickup. Speeds of all power turbines, except those that are decoupled, are indicated from this location. With the individual-governor approach, the control hardware would be increased by virtue of the extra power turbine governors. This would also complicate the gearbox and transmission system because of the three (or four) mounting locations and gear drives required.

For the free-turbine design, with the power turbine and gas producer constructed as an integral assembly, the power turbine speed could be sensed on each engine. A power turbine accessory drive must be provided for a lubricating pump for the turbine rotor; the power turbine governor can be driven from the same accessory drive system. The individual-governor approach would result in engines that include the complete gas turbine and control system intact. This feature would provide certain benefits in engine testing and field replacement. The single-governor approach could be employed, but the governor would have to be mounted on the combining gearbox (between helicopter rotor and overrunning clutches) to provide proper operation with any decoupled engines. The governor signal would have to be transmitted over a somewhat larger distance than for a system with individual governors driven by the turbine accessory drives.

Another free-turbine configuration considered is one which would utilize a single power turbine mounted directly on the combining gearbox, with the gas flow from all gas producers being ducted into this turbine. This configuration would utilize the single-governor concept since there would be only one power turbine. The operational characteristics and considerations would be essentially the same as those for the multiturbine configuration with the single-governor concept. Air valves may be required to regulate the airflow path when operating with one or more gas producers shut down.

The fixed-turbine-engine control system requires that each individual fuel control sense engine speed for transient scheduling, and thus could be employed within the control for governing. The single-governor approach could be employed, but similar to the integral assembly free turbine, the governor would have to be mounted on the combining gearbox. The governor signal would then have to be transmitted and employed in the fuel controls to effect the fuel metering. In this case, the single-governor approach may require more hardware components than the individual-governor approach.

Another system consideration is the ability for varying the governed speed of the helicopter rotor and/or trimming the governor setting to provide truly constant speed governing. With a single governor, the input signal, whether a mechanical throttle lever or electrical governor reset, can be transmitted to a single component. This eliminates the problems associated with unequal reset of governors in an individual governor control system. Also, location of the single governor on the combining gearbox should simplify system mechanization.

If constant rotor speed operation is required, the system must employ either an isochronous governor or a coordinated collective pitch-governor reset system (discussed in detail in a later subsection). In both concepts, the single-governor approach would be more suitable than individual governors. For the isochronous governor approach, it would enable simpler mechanization. For the coordinated system, the scheduling accuracy would be greater and the mechanization simpler.

Governor Failure Considerations

With the individual governor approach, the loss or failure of one of the governors such that the associated engine power is decreased to zero would not detrimentally affect the availability of the power of the other engines. Instead, the power of the remaining engines would automatically be increased by their governors as necessary to support the load. In the single-governor approach, the failure of this device could result in a power reduction in all engines. There would be no means available for recovering the power unless governor operation could be disarmed by the pilot, or a manual control provided.

If governor failure calls for maximum power (in the individual governor system), the other engines would be automatically controlled by their governors, reducing the power level as required to support the load. Should the load be less than the power available from the single engine

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operating at maximum power, governing could be overridden by the pilot-controlled throttle lever. The pilot could modulate down to ground idle (essentially zero power) by "twist-grip" operation of the fuel control throttle levers, thus manually controlling the rotor speed. This type of manual power control could also be employed in a single governor system with a governor failure that selects maximum power.

Load Sharing

Load sharing refers to the balancing of the engine powers in a clustered configuration so that the load is equally shared. In a clustered engine configuration with the output shafts mechanically connected so that they operate at identical speeds, load sharing can be accomplished either by open-loop or closed-loop control. The purpose of this subsection is to present an evaluation of the two concepts.

It is characteristic of gas turbine engines that their power and turbine temperature relationships vary somewhat in production. If the engines are controlled to operate at identical powers, they will not necessarily be at the same turbine temperatures (or gas producer speeds with a free-turbine engine). This characteristic must be recognized when evaluating load sharing.

Open-loop load sharing relies on the accuracy with which engine performance characteristics can be predetermined and the control system prescheduled. Closed-loop load sharing would entail sensing the performance of each engine, comparing performance to a "master," and trimming the controls of the "slave" engine so that they are sharing the load equally.

Open Loop

The open-loop evaluation was directed towards establishing the accuracy of matching that could be accomplished, considering engine and control variations in production. Allison production test data were utilized in establishing the gas turbine engine variations in production. The data accumulated relative to the T56 and T63 production engine performance variations indicate that the engines possess a similar power variation in production. For the T56 this is a maximum of 10-percent-of-point total variation in power level at a specific fuel flow. For the T63 this is a maximum of 10-percent-of-point total variation in power level at a specific gas producer speed (see Figure 56).
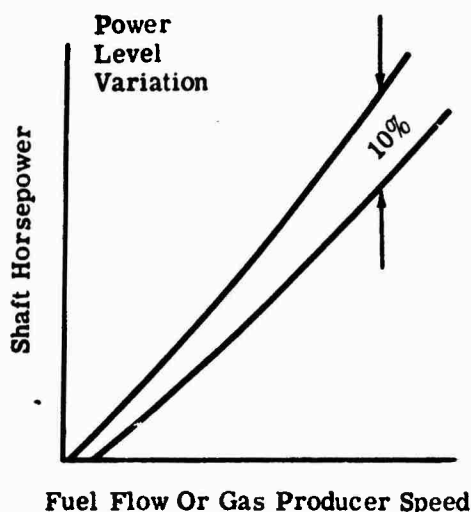
Figure 56. Power Level Variation
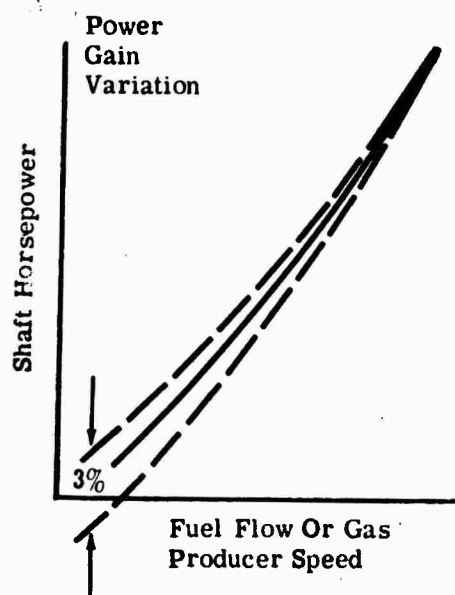
Figure 57. Power Gain Variation

The variation in the gains of power to fuel flow and gas producer speed was found to be small, being equivalent to approximately plus or minus 1.5 percent of maximum power (around nominal) in the minimum power range. See Figure 57. This is true for both the T56 and T63 engines.

The control-system production variations were defined previously. The engine and control-system production variations must be combined in arriving at the potential total variations that could be experienced with open-loop load sharing. The accuracy can be enhanced by special system consideration. For example, the control-system variations would be greatly reduced by the single-governor control system, as compared to individual governors. This is true for both the fixed- and free-turbine engines. Also, the power level variations between engines could be adjusted in the rigging of the engines in the helicopter.

The tabulation presented in Table 10 indicates the maximum potential variation in power between engines with fixed and free turbines and with single and individual governors. These data include production engine

TABLE 10
MAXIMUM POTENTIAL VARIATION IN POWER BETWEEN
ENGINES WITH FIXED AND FREE TURBINES AND
WITH SINGLE AND INDIVIDUAL GOVERNORS

| Engine Design | Power Level | Total Power Variation (% of Maximum) | |
		Single Governor	Individual Governor
Fixed Turbine	Maximum	5.7	5.0
Fixed Turbine	Minimum	8.7	26.0
Free Turbine	Maximum	5.0	10.0
Free Turbine	Minimum	8.0	30.5

gain variations, but not differences in power levels. These data indicate that the power unbalance could be minimized by selection of the single-governor control concept over that which would be obtained with individual governors.

Closed Loop

Closed-loop load sharing could be accomplished by sensing the engine torques, comparing the levels in a load-sharing computer, and utilizing the signals generated by the computer to trim the governor settings of the appropriate fuel control. The conventional approach is to maintain one engine as a master, operating the other engines as slaves. The shaft torque of each engine would be sensed and transmitted to an electronic load-sharing control. In this control, the torque signal of each slave engine is compared with the signal from the master engine. The error signals generated would be transmitted to the appropriate fuel controls to trim the fuel scheduling, thereby matching the power output of the slave engines to the master.

Load sharing would be mechanically limited in the fuel controls to protect against electronic control failures. In case electronic control failure did occur, it would have the effect of slightly changing the governor speed setting (Figure 58). This would not prevent being able to obtain maximum power from the power system, nor would it prevent obtaining zero power. The effect would be that a somewhat larger rotor speed droop would be experienced in modulating power from zero to maximum. This is illustrated by Figure 59, which assumes one control system reset to the full lean condition. The increase in governor droop is equal to effective reset of the governor (caused by the control failure).

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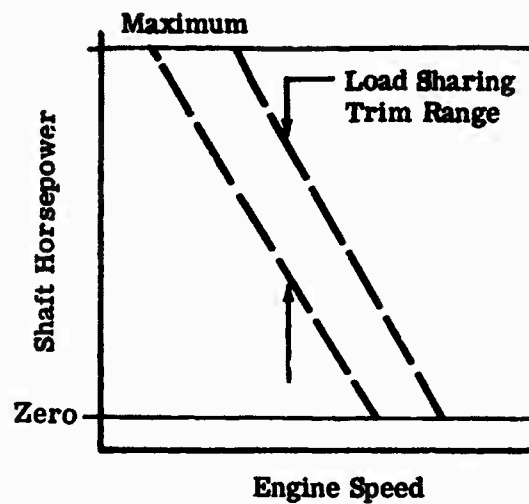
Figure 58.   Load Sharing Trim Range
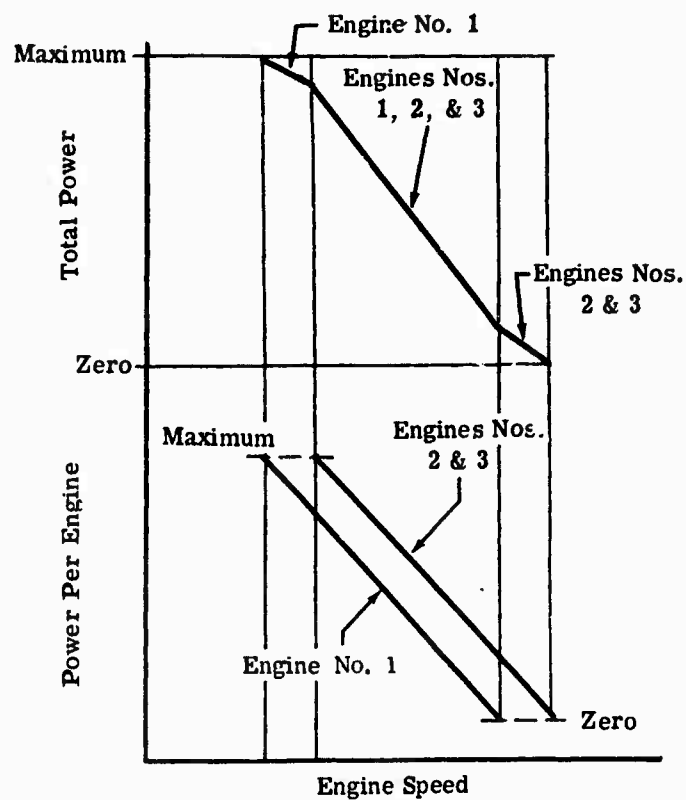
Figure 59.   Effect of Full Lean Governor Setting of One Engine

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A similar result will occur if the master engine has a low power output versus turbine temperature characteristic. The turbine temperature limiter would automatically limit the output of this engine, and the load-sharing control would attempt to prevent operation of the other engines at any power level greater than the master engine. The result would be that the other engines would have their fuel controls reduced toward the minimum setting (maximum reset lean), effectively setting their governors to a lower speed setting. Figure 60 illustrates that a rotor speed dead-band region would occur until the fuel control mechanical reset limit prevents further fuel control leaning.

The response of the load-sharing control should be slow to prevent it from destabilizing governing operation. The power transient response of the engines would not be dependent on the response of the load sharing function, for it would not affect the availability of maximum and minimum power.

The free-turbine configuration employing a single power turbine, gas-coupled to multiple gas producers, would require sensing some parameter other than shaft torque for closed-loop load sharing. One approach would be to match the turbine temperatures of the engines by an automatic control.
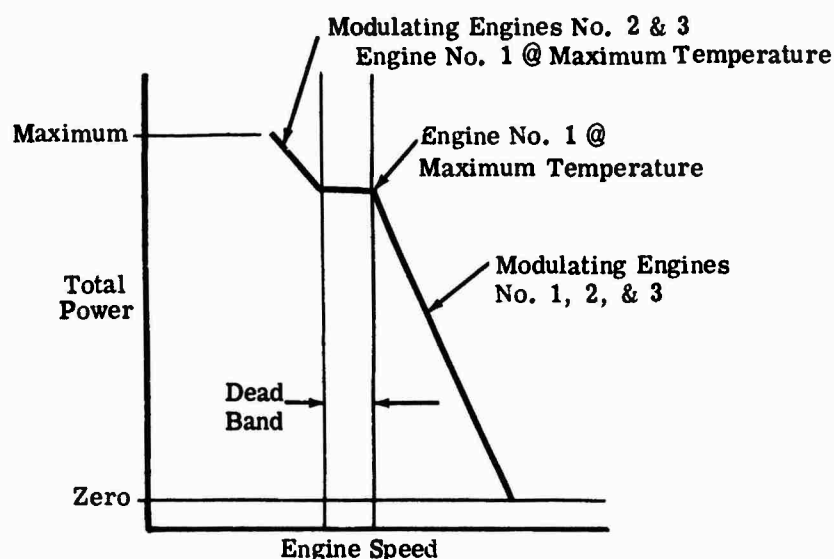
Figure 60. Effect of Load Sharing Master Being Low in Power Output

Isochronous Governing

A requirement of the HLH engine control system is that automatic rotor speed governing be provided, with the engine controls regulating the power according to the load. A conventional, proportional (droop) governor would require a small variation of engine and rotor speed to match the rotor load, with the gain of the governor being established to provide stable operation (Figure 61). The rotor speed would then vary with the power level and ambient conditions. Constant rotor speed operation is desirable, independent of power level and ambient conditions. This can be accomplished by open-loop or closed-loop methods.

Coordinated Collective Governor Levers

One open-loop concept that is widely employed involves resetting the governor by the collective pitch according to a predetermined schedule (see Figure 62). This approach would allow trimming out the droop of
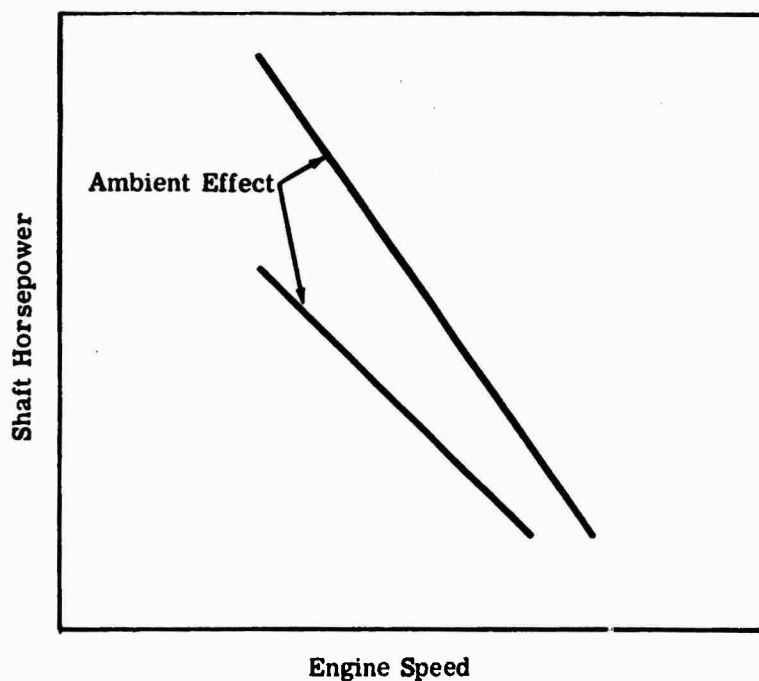
Figure 61. Available Horsepower With a Proportional Governor

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the governor, but would not compensate for the effect of ambient conditions or airspeed. Also, the predetermined schedule must be based on all engines operating, and it would result in governing at a lower speed when any engine is shut down (Figure 63). The reason for the speed variation is that neither the governor lever setting nor the collective pitch positions are true indicators of available power or load.

The coordinated lever scheme introduces a desirable lead into the system on load transienic. The change in collective pitch initiates a fuel flow change before a speed change occurs, thus tending to minimize transient rotor speed variation. As was previously indicated, in this scheme the collective pitch control lever would be connected to the governor speed-setting lever. For the individual-governor concept, the collective pitch control must also reset the levers on all controls, thus imposing another schedule variation relative to load sharing. The production variations of governor setting as a function of governor lever would be additive to the previously defined control tolerance. The single-governor approach would involve simpler mechanisms and would not include governor lever-governor setting production variations in the load sharing picture.
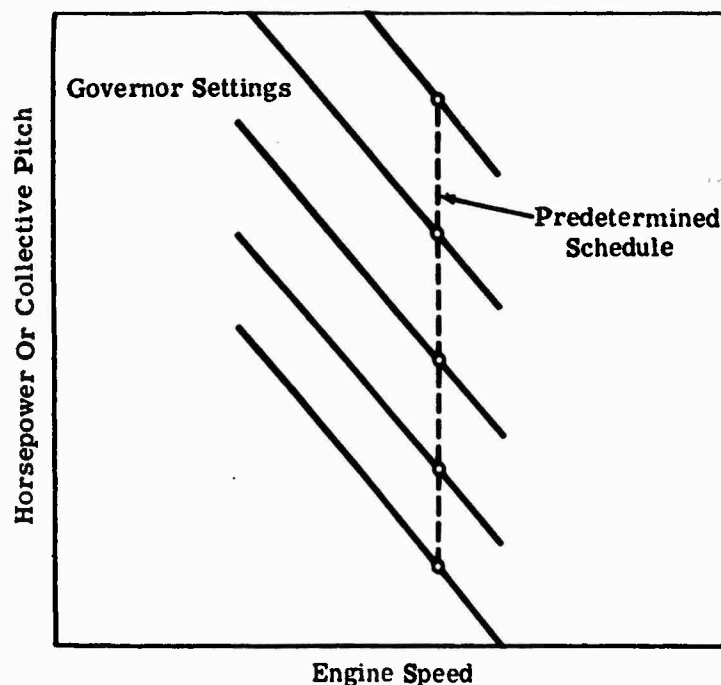
Figure 62. Collective Pitch and Governor Lever Coordinated Scheduling
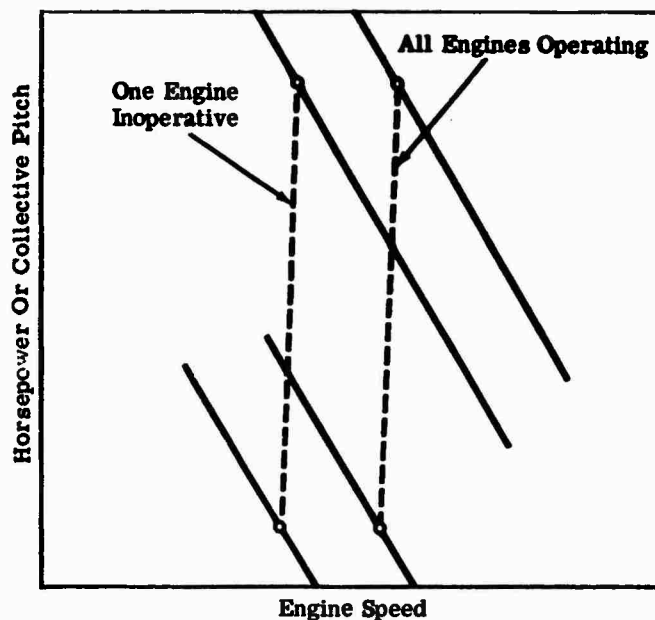
Figure 63. Speed Shift With One Engine Inoperative and Coordinated

Isochronous Governor

Closed-loop, isochronous governing could be achieved by a proportional-plus-integral reset governor. This type of control would utilize a fast responding proportional governor to provide the power transient characteristics and stability required, with a slow response integrating reset to eliminate any speed error. By employing the speed error signal to continue to reset the proportional governor until the error is eliminated, the rotor speed would be maintained constant for all steady-state operation. The effects of ambients, airspeed, and/or single-engine shutdown would be compensated for automatically.

The isochronous feature of the governor would lag during power transients. The fuel flow would be regulated by the proportional governor, requiring a speed change before a fuel flow change would be initiated. Thus, the isochronous governor would not possess the lead feature provided by the coordinated approach. The result would be transient rotor speed variations greater than those with the coordinated approach.

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The authority of the integrating reset portion of the governor must be limited to provide protection against incorrect operation of this mechanism. The operation for a failed integrating loop in the governor would be similar to that described for a load-sharing control failure, allowing full power operation, but at a slightly incorrect speed.

With the individual-governor concept, all engines cannot employ isochronous governors. This is because of the operating characteristics that would occur with a slight error in the speed setting of the integrating portion of the controls. Operation of this type of system would be unacceptable because it would result in extreme power unbalance between engines. Figure 64 illustrates the governing characteristic that would result over the power range.
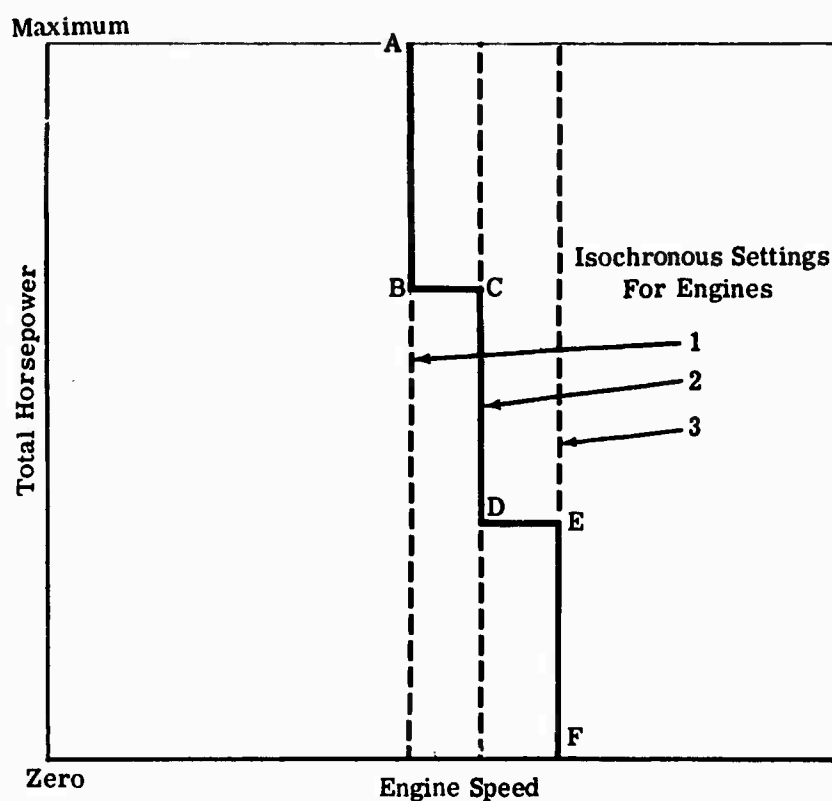
Figure 64. Potential Characteristics With Individual Isochronous Governors

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Table 11 defines the engine operating conditions throughout the power range, using three isochronous governors.

TABLE 11
ENGINE OPERATING CONDITIONS THROUGHOUT POWER RANGE
USING THREE ISOCHRONOUS GOVERNORS

Points	Engine 1	Engine 2	Engine 3
A	Temp. Limited	Temp. Limited	Temp. Limited
A to B	Modulating Power	Temp. Limited	Temp. Limited
B to C	Zero Power	Temp. Limited	Temp. Limited
C to D	Zero Power	Modulating Power	Temp. Limited
D to E	Zero Power	Zero Power	Temp. Limited
E to F	Zero Power	Zero Power	Modulating Power
F	Zero Power	Zero Power	Zero Power

The operation of an engine at the turbine temperature limit results from the speed being less than its isochronous governor setting. The operation at zero power (decoupled) results from the engine speed being greater than the governor setting.

The conventional approach is to use an isochronous governor on one engine, with the other engines employing proportional governors. To vary the power generated by the "proportional-governor-controlled" engines, these governors must be reset according to the load. This could be accomplished by employing a closed-loop load sharing scheme with the isochronous governor on the master engine and the slave engines controlled by the proportional governor.

The single-governor concept (one governor to operate all fuel controls) could readily utilize an isochronous governor and would not require closed-loop load sharing. Because it involves only a single governor, it would not be subject to the dynamic incompatibility that would occur with multiple isochronous governors.

Control Systems

As is indicated in the preceding subsections, there are various control system concepts that can be employed on the HLH powerplant. Each scheme has certain advantages and disadvantages. This subsection presents two different concepts to illustrate the system integration required between the helicopter and the individual engines.

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Figure 65 illustrates a typical control system employing the single-governor concept. This system would be applicable to either free- or fixed-turbine engines. An isochronous governor (proportional with integral reset) could be utilized to maintain constant rotor speed operation in steady state. A pilot-controlled governor "beep" signal is illustrated for trimming or resetting the rotor speed governor as desired to improve rotor system efficiency. The single governor generates and transmits a reset signal (probably hydraulic) to all three engine controls to affect the fuel flow for governing.

The individual fuel controls incorporate a control lever that is mechanically connected to a pilot control point. The helicopter "twist grip" on the collective lever would operate the mechanical inputs of the control, providing the three positions of "off," "ground idle," and "run." The "off" position would mechanically close the fuel cutoff valves in the individual controls. In the "ground idle" position, the controls would directly regulate fuel flow, providing a ground idle rotor speed. During this operation, the isochronous governor would not be controlling fuel flow. In the "run" position, the scheduling of the individual controls is set for maximum power, with the rotor speed governor reducing the fuel flow as required to match the load. The isochronous governor provides power control. Manual power modulation could be accomplished in the event of a governor failure that selected maximum power. By modulating the "twist grip" between "ground idle" and "run," the pilot could manually control the power level and rotor speed.

Individual electrical switches (or manual devices) operating fuel cutoff valves in the controls could be employed to enable shutdown of individual engines for cruise economy, or under an emergency condition. These fuel cutoff switches would also be tied into the automatic starting control systems of the engines.

The individual fuel controls would utilize the following engine parameters to properly schedule the fuel flow and vary engine compressor geometry: compressor inlet air temperature (CIT), compressor discharge pressure (CDP), and engine speed (N_E). A turbine inlet temperature (TIT) signal would be employed to provide closed-loop turbine temperature limiting.

Figure 66 illustrates a control system employing individual governors and closed-loop load sharing. The governors are proportional, employing coordination of the collective and governor levers to trim out the droop. The scheme as illustrated is for a free-turbine engine, but

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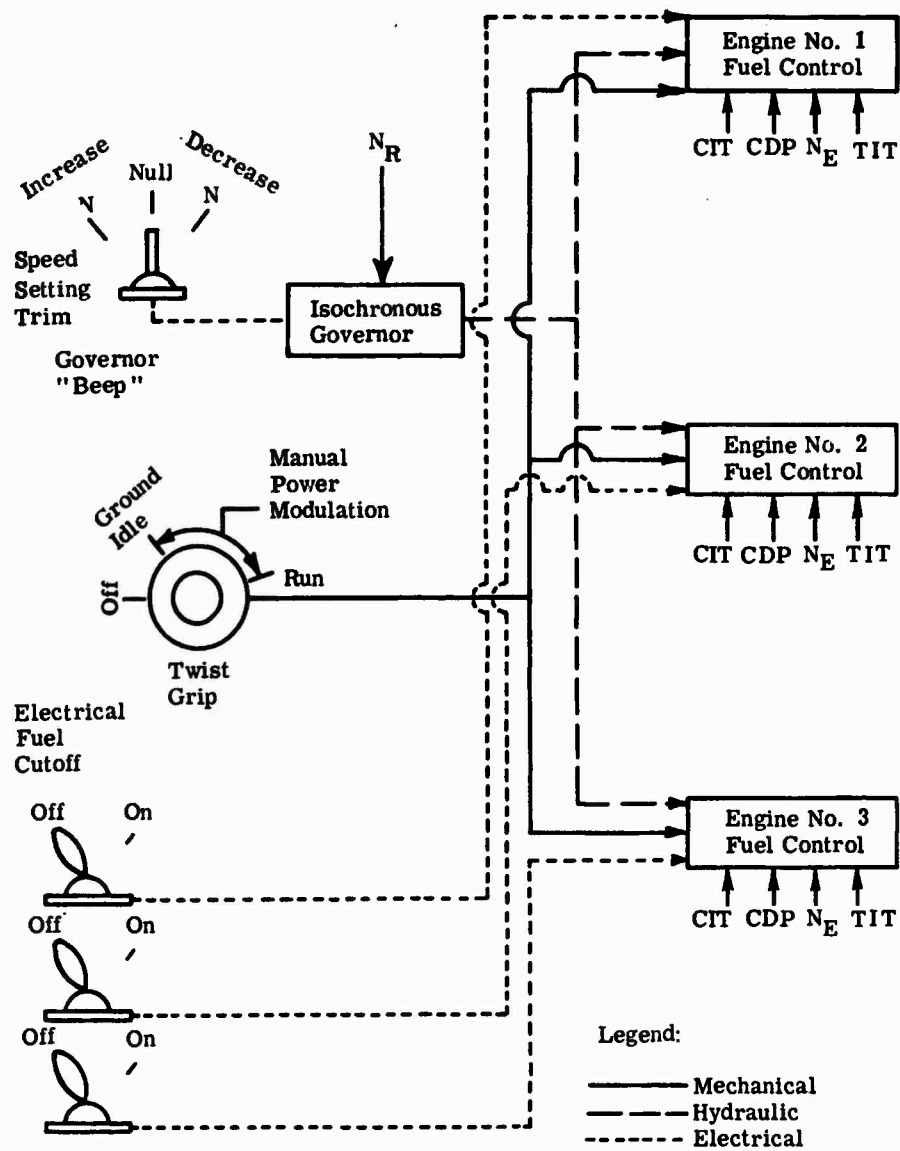
Figure 65. Single-Governor (Isochronous) Control System

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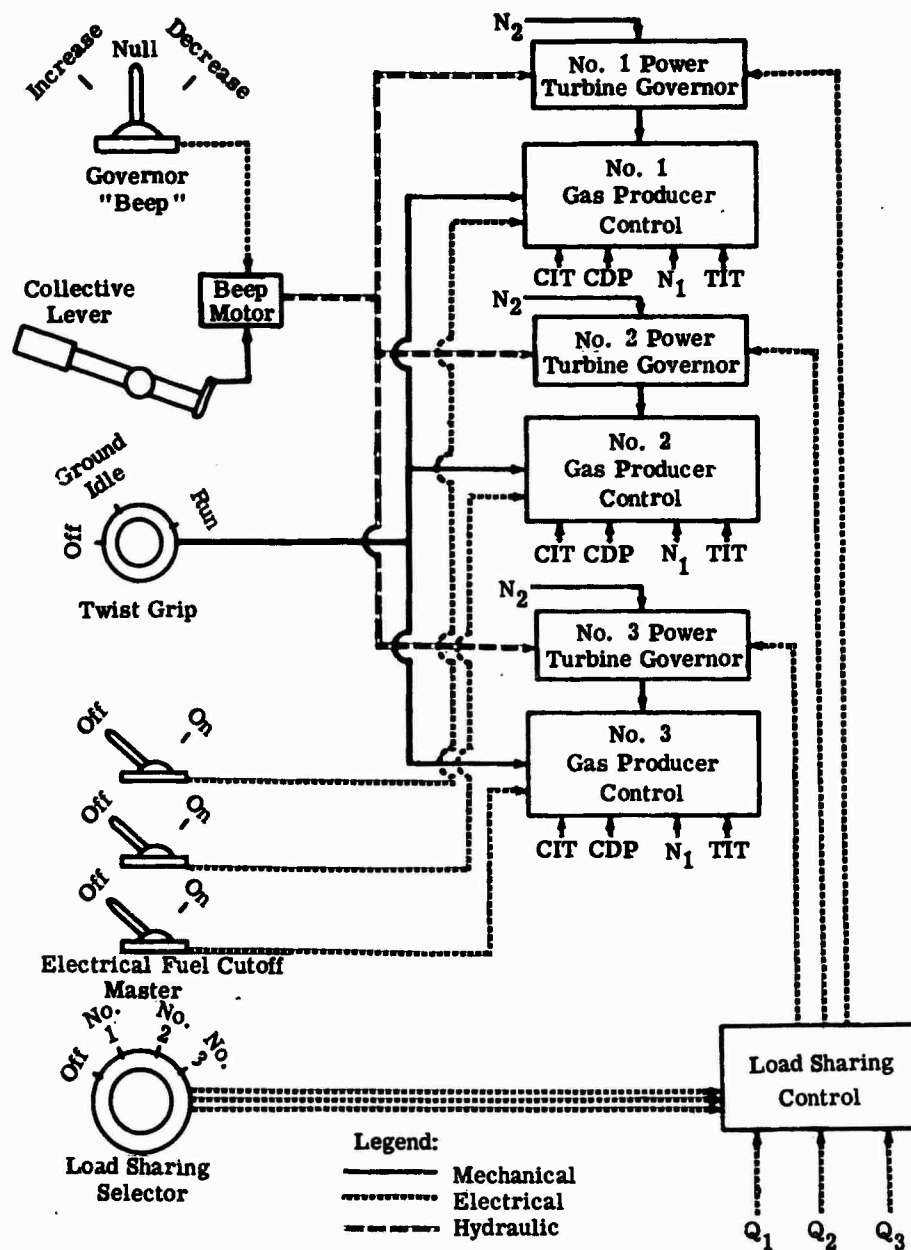
Figure 66. Individual-Governors (Coordinated With Closed-Loop Load Sharing) Control System

the concept is directly applicable to the fixed-turbine engine. The governor levers are interlinked and mechanically connected to the collective pitch lever to enable coordination. The governor "beep" switch operates a motor in the mechanical linkage system to effectively reset the speed setting as desired for optimum rotor efficiency.

Operation of the fuel control levers by the "twist grip" would be the same as for the system described previously. The load sharing selector switch would be required to activate the control and to select a specific engine for a master. The load sharing control would receive torque signals from each engine and transmit electrical signals to the individual control systems (except the master) to provide a trimming reset. The fuel controls would utilize parameters similar to the single governor system (Figure 66). Each fuel control would employ a gas producer speed (N_1) signal. Each governor would employ a power turbine speed (N_2) sense.

DYNAMIC ANALYSES

The dynamic studies are related to potential problems and special requirements associated with the large rotor system and/or clustered engine operation. The analyses were directed toward engine power transients and rotor system torsional stability.

Power Transients

The power transient studies were conducted to establish the engine power response capability required to be compatible with the large helicopter. The transients of concern are: rapid load transients introduced by movement of the collective pitch and power transients initiated by an abrupt loss of power from one of the engines.

Lift or thrust from the rotor system is a function of the rotor speed and the angle of the blades (collective pitch). The pilot, when requiring a high lifting force, increases the collective pitch and the engine(s) must provide the power to maintain rotor speed. Lag in the engine power response on a rapid collective increase results in undesirable transient rotor speed droop and associated lower rotor lift during the transient.

Figure 67 illustrates that an engine power ramp response time of 3 seconds (or faster) is required to prevent a transient rotor speed droop of greater than 10 percent. This is based on a load increase from zero

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power to two-thirds of the maximum available system power, with the load being applied uniformly in a 1-second interval. A 1-second engine response characteristic would theoretically result in zero transient speed droop, assuming no delay between the power demand and power availability. Two-thirds of the maximum total power was assumed as the maximum load, based on a three-engine configuration and with the assumption that the powerplant must be capable of supporting the load in the event of the loss of one engine.

The data presented by Figure 67, based on a free turbine engine, indicate that the time to complete the gas producer rotor speed change from minimum to maximum must be 3-seconds or less. Also, if any variable geometry is utilized, its positioning must be accomplished in this period of time from the instant of initiation of the load increase.

Although the data of Figure 67 are for a free-turbine engine, they are also applicable to fixed-turbine engines. The fixed-turbine engine in a helicopter would be maintained at a constant speed in normal operation. Power changes would therefore be accomplished by changing the fuel
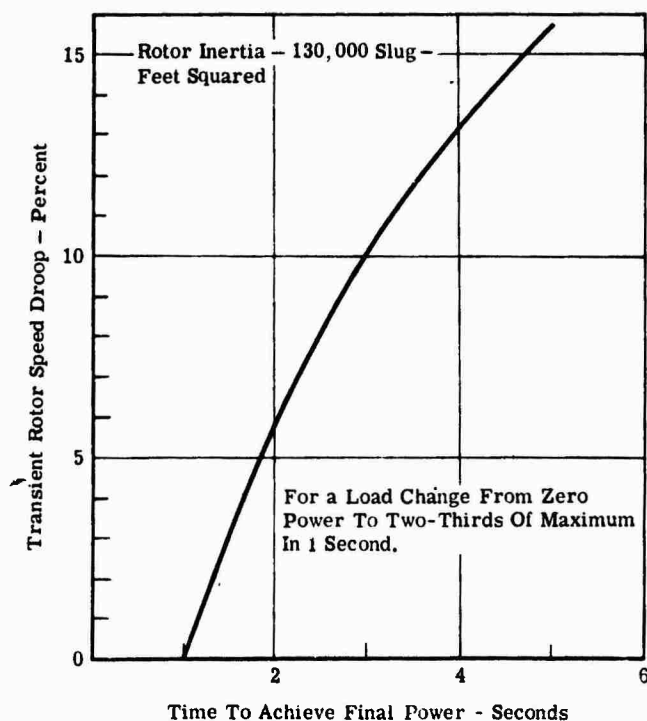
Figure 67. Effect of Engine Power Response on Transient Rotor Speed Droop

flow (and in some cases variable geometry) and would not require an engine speed change with its associated lag. The response of the fuel control and positioning system would be 1 second or less for a 100-percent flow or position change, with a delay (caused by control lag) of less than 1/2 second. The result would be a maximum transient rotor speed droop of less than 5 percent.

The interruption of the fuel flow to an engine would result in a nearly instantaneous loss of power from that engine. In a clustered configuration, the remaining engines must pick up the load by increasing their power levels to provide the total shaft power required. Rapid power response is required in a critical hover condition where the lift must be maintained constant to prevent a change in altitude.

The lag in engine power buildup, with the occurrence of any engine failure, would result in some rotor speed reduction. At a fixed collective pitch this speed reduction would result in a decrease in rotor lift. To prevent the decrease in lift, the collective pitch must be increased. Associated with the increased pitch is an increase in power required to drive the rotor (Figure 68), further tending to reduce rotor speed. However, the large rotor system possesses kinetic energy that can be
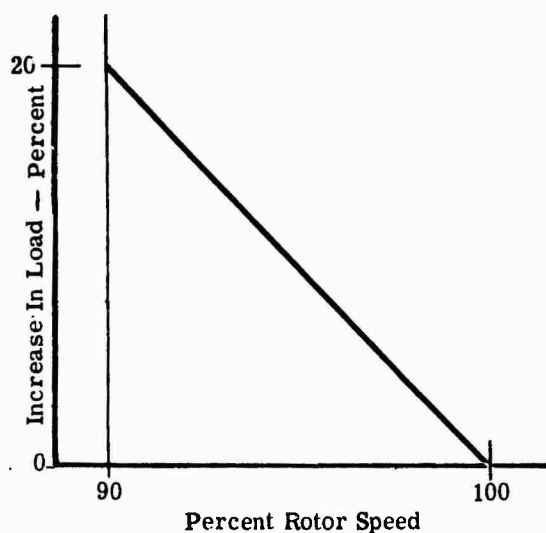
Figure 68. Rotor Speed-Load Characteristic for Constant Rotor Lift

utilized to help support the load prior to the completion of the engine power buildup. With a three-engine system, the absolute limit of the maximum load which could be supported with one engine failure would be that which requires 66.7 percent of the maximum installed power, or 100-percent power from the two remaining engines. Upon the loss of power from one engine, this condition would require an instantaneous increase to maximum power on the two engines operating (resulting in zero rotor speed droop).

In actuality there would be some lag involved in the power buildup and the rotor speed would droop below 100 percent. Figure 69 defines the relationship between engine power response required for zero change in lift, as a function of the initial steady-state power level (load), and transient rotor speed droop. (It should be noted that a linear power-time characteristic was assumed.) The response time is from initial power level to maximum for the two operational engines. For a load which initially requires 50 percent of maximum power, with three engines operating, the engine response time capability must be 3 seconds to allow zero change in lift and limit the transient speed droop to 5 percent (point A). If the response time is 5 seconds, the transient speed droop will be 10 percent, for zero change in lift (point B).

Should the collective pitch be increased at a rate to cause a 10-percent speed droop, with an engine response of 3 seconds and a 50-percent load, the lift would be greater than the initial lift and the helicopter would climb. Conversely, if the pitch change rate is restricted to limit the transient droop, the lift will be less than initially and will result in a loss in altitude.

Any combination of initial power and engine response time which falls to the right of the dashed line on Figure 69 will result in a loss of lift and continued rotor droop. This condition results from the engine's being unable to supply the increased power required to maintain constant lift at reduced rotor speed. (See Figure 68.) Recovery from this possible region of transient operation would require unloading of the rotor.

Torsional Stability Analysis

The purpose of this phase of the study was to estimate the degree of stability of the rotating portion of an HLH propulsion system. Knowledge and experience gained from light helicopter engine development
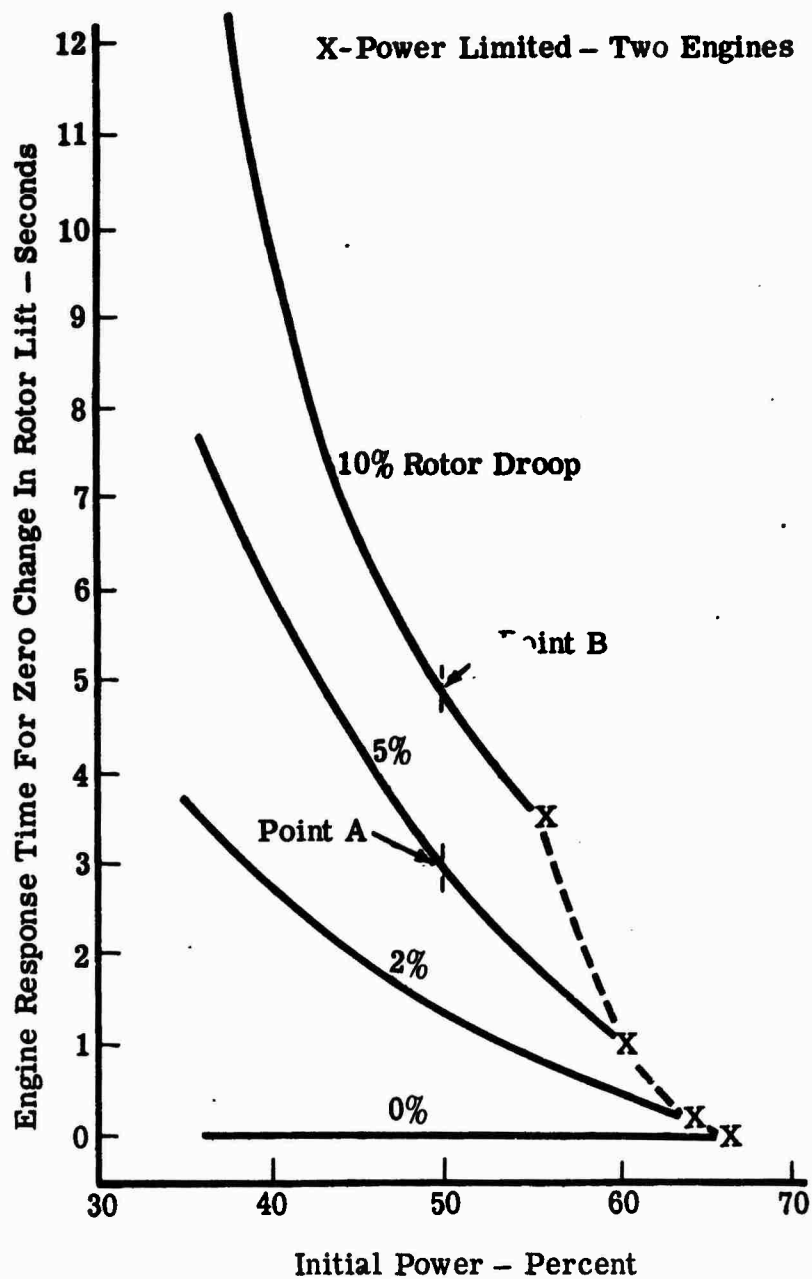
Figure 69. Engine Power Response Required for Zero Change in Rotor Lift

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were used in studying the more complex multiengine propulsion system for the HLH. Both fixed and free turbine engines (with and without regenerator) and both single and tandem rotor systems (with and without drag hinge) were investigated. Stability analyses were coordinated with the results of control mode studies and engine performance studies.

The analog computer was used extensively in this work, since this type of problem is ideally suited to linearized analog simulation and can be effectively studied in this manner. Whenever possible, realistic data pertaining to the rotor were obtained from airframe manufacturers, while engine performance and fuel control data were obtained from other Allison groups involved in this study. Where specific data were lacking, approximate values were obtained by scaling up other existing system parameters. Generalized studies were made which permit stability predictions as a function of the design-controllable variables.

Potential Problem Areas

Experience with smaller helicopter propulsion system designs has confirmed the necessity for careful analysis of the system with regard to torsional stability. The rotating portion of a turbine-powered helicopter (Figure 70) consists basically of several inertias interconnected with shafts having some torsional resiliency; i.e., a finite stiffness. Some inherent damping is associated with each inertia.

It would be desirable to sense and utilize rotor speed as a control parameter, however, it is not practical to measure this quantity directly. Instead, engine output shaft speed is sensed and fed back to the fuel control which adjusts fuel flow to maintain the desired speed. Since rotor inertia is much larger than turbine inertia, rotor speed rarely follows torsional oscillations. However, the shaft and turbine rotor form a highly underdamped system which will tend to oscillate at a frequency of around three to five cycles per second for a large helicopter. If the gain and the dynamics (lag time constants) of the fuel control are such as to reinforce these oscillations, they may become divergent. The most straightforward way to prevent this is to raise the torsional natural frequency by increasing shaft stiffness or decreasing turbine inertia. This will always require a compromise, since the former imposes a weight penalty and the latter is limited by engine design. Likewise, reduction of control system lags, or the addition of compensation devices, may become unrealistic from a design viewpoint—these factors also must be considered in the system compromises. The results of this study should aid in making these compromises.

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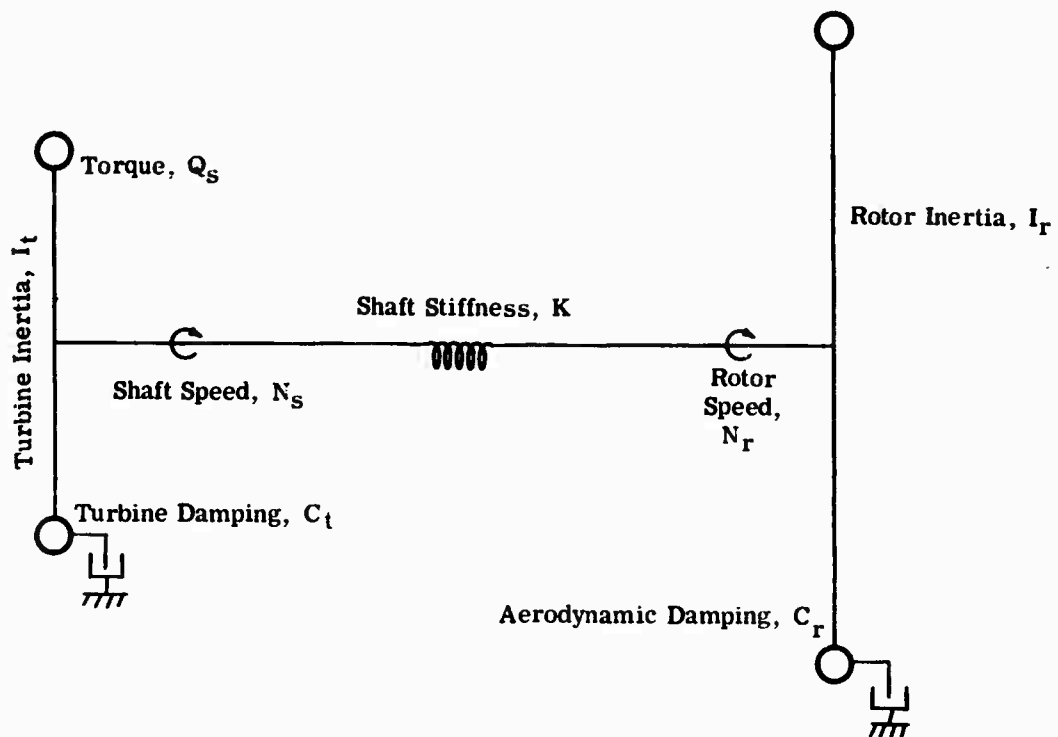
Figure 70. Engine-Simplified Rotor System Schematic

Fixed Turbine Systems

An analog computer simulation was made of a powerplant consisting of four T56 engines, a rotor system in which the only known parameters were rotor inertia and speed, and one speed governor controlling fuel flow to each engine.

The system simulated is shown in block diagram form in Figure 71. The purpose of the study was to determine the various rotor, engine, and fuel-control parameters which would most strongly influence torsional stability. The simulation is linearized around the 100-percent speed, full-power point. Experience has shown that torsional oscillations are most likely to occur at full power.
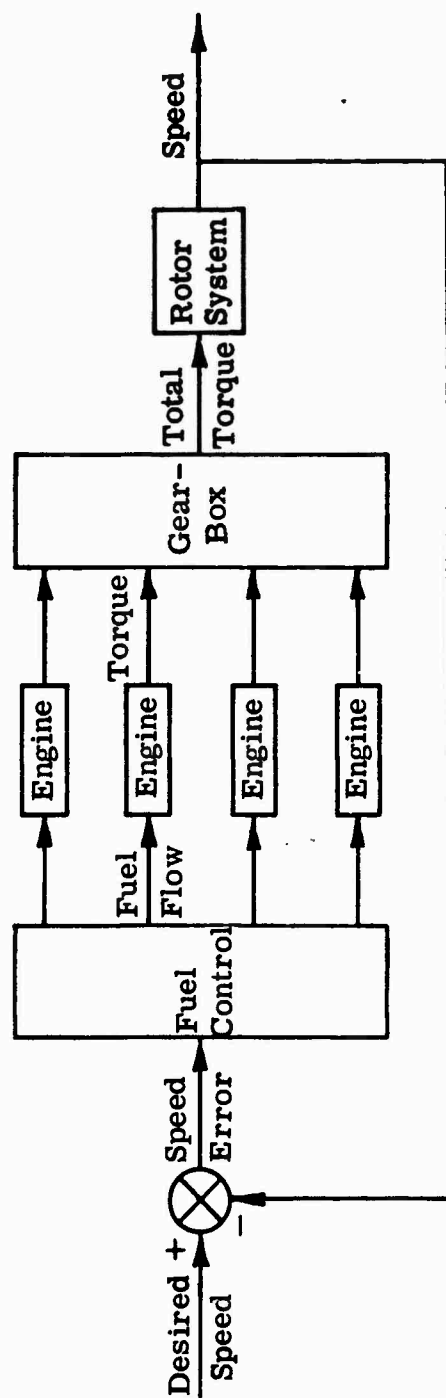
Notes:

1. Four T56 engines are assumed to have identical characteristics.

2. Fuel control contains two first-order lags of 0.1 sec each, zero hysteresis, and gain corresponding to 4% governor droop.

3. Rotor inertia = 15.3 slug ft^2 referred to 100% engine speed, 13820 rpm. All speeds and torques referred to this speed.

Figure 71. Multiengine Helicopter Powerplant Block Diagram

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Figure 72 is a block diagram of the T56 engine and fuel control as they were simulated on the computer. Partial derivatives were obtained from T56 engine maps. The multiengine cluster was simulated by multiplying the torque output of one engine by four, the number of engines. Values used in the simulation are as follows:

K_G = 1.81 pounds per hour per gas producer r.p.m.

τ_1 = 0.1 second

τ_2 = 0.1 second

		Hot Day	Standard Day	Cold Day
$\dfrac{\partial T_{T5}}{\partial W_f}$,	$\dfrac{\text{degree F.}}{\text{pounds per hour}}$	0.5	0.5	0.5
$\dfrac{\partial Q_e}{\partial T_{T5}}$,	$\dfrac{\text{pound-feet}}{\text{degree F.}}$	1.8	1.8	1.8
$\dfrac{\partial T_{T5}}{\partial N_e}$,	$\dfrac{\text{degree F.}}{\text{r.p.m.}}$	0.12	0.035	-0.01
$\dfrac{\partial Q_e}{\partial N_e}$,	$\dfrac{\text{pound-feet}}{\text{r.p.m.}}$	0.023	0.100	0.160

Figure 73 shows the results of the study. It defines regions of stability and instability for various combinations of engine inertia and shaft stiffness. It indicates that stability problems will be greatest on a cold day and when fuel control lags are around 0.06 second. Assuming more favorable fuel control lags of 0.1 second and an engine inertia of 1.71 slug-feet squared, shaft stiffness should exceed 3500 pound-feet per radian for torsional stability. For a given rotor inertia, the natural frequency of the engine-shaft-rotor system is determined by engine inertia and shaft stiffness. This frequency must be greater than 4.5 cycles per second for stability on a cold day, and 3.5 cycles per second on a standard day.

Increasing governor droop to 5 percent, or introducing a small amount of hysteresis in the governor, eliminated oscillations for all combinations of inertia and stiffness. Fuel control lags of 0.06 second appear to be a "worst-case" condition, but this is easily avoided in fuel control design.
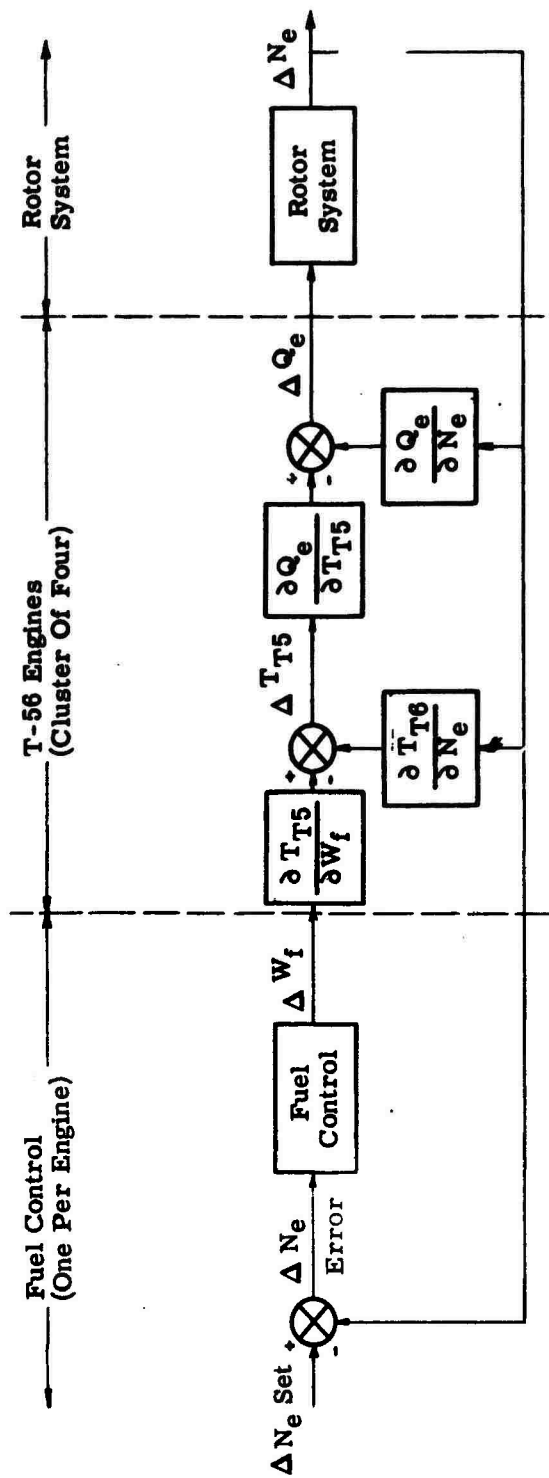
Fuel Control Transfer Function:

$$\frac{\Delta W_f}{\Delta N_e \text{ Error}} = \frac{K_G}{(\tau_1 S + 1)(\tau_2 S + 1)} \quad \text{(Hysteresis May Be Added)}$$

Figure 72. Model T56 Engine and Fuel Control Block Diagram
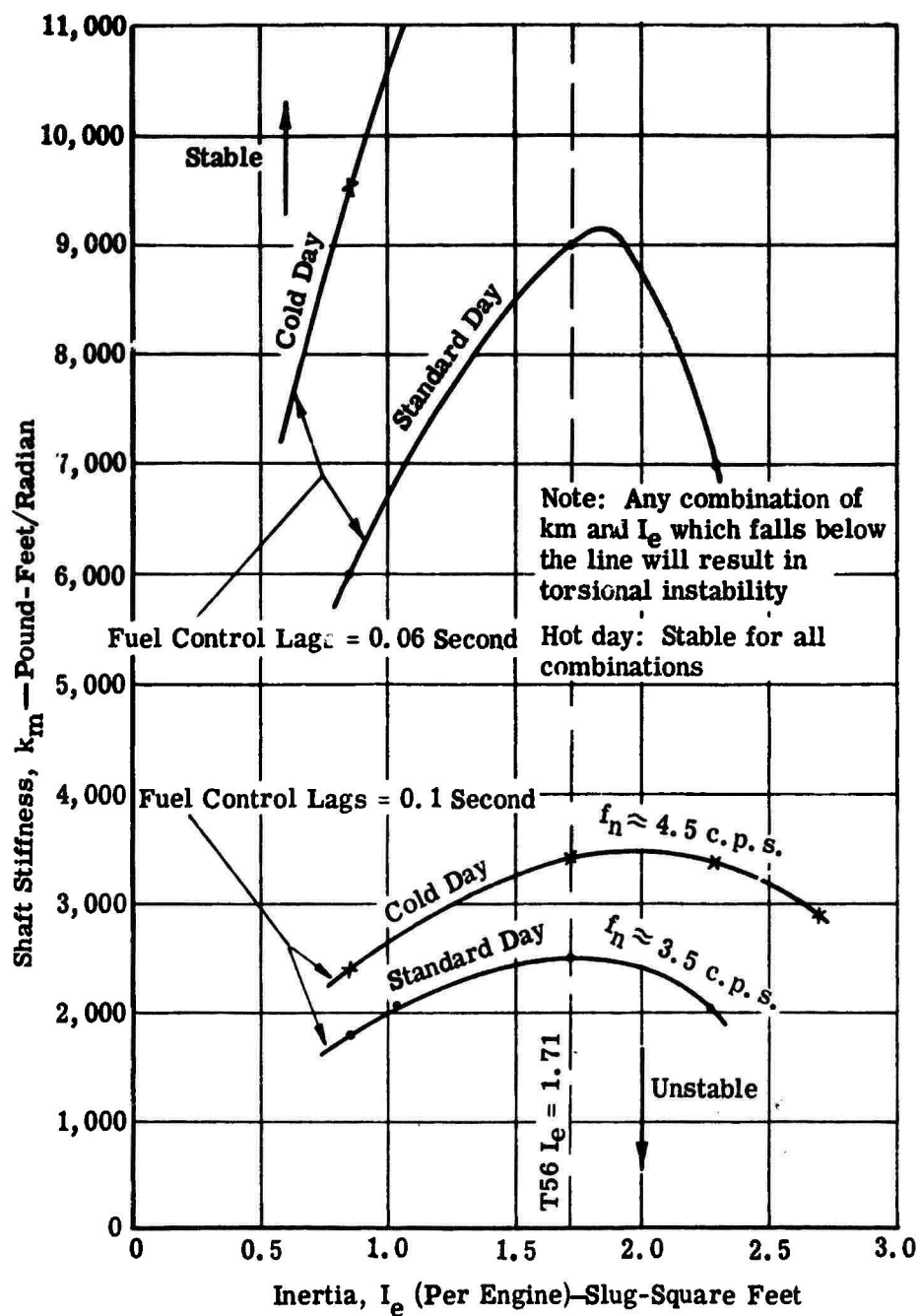
Figure 73. HLH Torsional Stability Analog Computer
Data—Four Fixed-Turbine Engines

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The analog computer simulation of four T56 engines driving a rotor system was modified to include a regenerator on each engine. The purpose of this phase of the study was to determine the effect on torsional stability of using fixed turbine engines with regenerators.

A flow diagram with station numbers is shown in Figure 74.
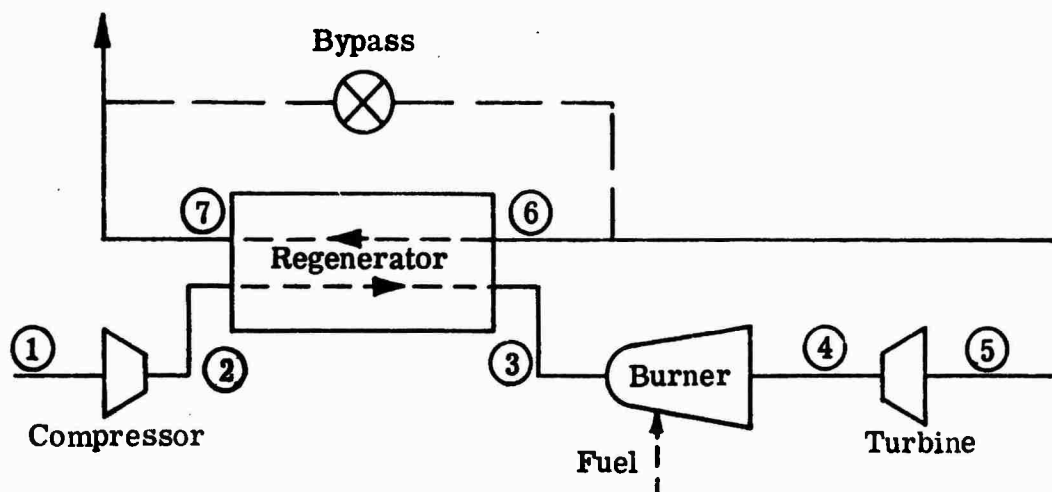
Notes:
1. $T_4 = T_3 + \Delta T$ where ΔT (burner rise) is assumed to vary linearly with fuel-air ratio

2. $T_3 = \dfrac{T_6 \, \epsilon}{(\tau_6 S + 1)} + \dfrac{T_2 (1-\epsilon) \, e^{-T_d S}}{(\tau_2 S + 1)}$

where, assuming high-flow, zero bypass:

ϵ (regenerator effectiveness) = 0.7
T_d (time delay) = 0
τ_6 = 0.5 sec
τ_2 = 1.0 sec
S = LaPlace transform operator

Figure 74. Flow Diagram for Fixed Turbine Engine
With Regenerator

The results of the study are shown in Figure 75. In general, adding the regenerator improves torsional stability. This is mainly due to the lag and attenuation between T_2 and T_3. Since T_2 increases with speed, this normally has a destabilizing effect. On standard day, with nominal lags and engine inertia, shaft stiffness, k_m, may be reduced from 2500 to 2200 pound-feet per radian when the regenerator is added.

Free Turbine Systems

The analog computer was used to study the torsional stability of a helicopter employing a cluster of three free turbine engines and several different rotor configurations.

Figure 76 shows the block diagram of the 548-C2 engine and fuel control as they were simulated on the computer. Partial derivatives describing engine performance were obtained from IBM program L14. The three engines were simulated separately and independently to allow for possible interaction and to permit studying the effect of the loss of one engine.

Gains and time constants used in the 548-C2 engine simulation are as indicated in the following tabulation (see Figure 76):

K_p = 3.75 gas producer r.p.m. per shaft r.p.m. (corresponds to a droop of approximately 4 percent)

K_G = 0.4 pound per hour per gas producer r.p.m.

K_{CDP} = 13 pound per hour per p.s.i.

τ_1 = 0.10 second

τ_2 = 0.10 second

τ_3 = 0.10 second

τ_4 = 0.07 second

τ_5 = 0.05 second

τ_6 = 0.02 second

τ_7 = 0.10 second

Analog Computer Data
Four Engines Clustered
Fixed Turbine With Regenerator
4% Governor Droop
Sea Level Static

Curve (1) Standard Day, Without Regeneration, 0.1-Second Fuel Control Lags

(2) Standard Day, With Regeneration, 0.1-Second Fuel Control Lags;
Regeneration Time Constants τ_2 = 1 Second, τ_6 = 0.5 Second

(3) Standard Day, With Regeneration, 0.1-Second Fuel Control Lags;
Regeneration Time Constants τ_2 = 2 Seconds, τ_6 = 1 Second

(4) Standard Day, With Regeneration, 0.06-Second Fuel Control Lags;
Regeneration Time Constants τ_2 = 1 Second, τ_6 = 0.5 Second

(5) Cold Day, Without Regeneration, 0.1-Second Fuel Control Lags

Note: Found To Be Stable For All Conditions of k_m And I_e

Cold-Day Operation With Regeneration

Hot-Day Operation

Operation With 5-Percent Governor Droop

Any Combination Of K_m And I_e Which Falls Below

The Line Will Result In Torsional Instability

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Stable

(4)

6000   5000   4000

nd-Feet/Radian

1

Cold-Day Operation With Regeneration

Hot-Day Operation

Operation With 5-Percent Governor Droop

Any Combination Of K_m And I_e Which Falls Below

The Line Will Result In Torsional Instability

Shaft Stiffness, K_m — Pound-Feet/Radian

Stable

Unstable

$(T-56)$ $I_e = 1.71$

Inertia, I_e, (Per Engine) — Slug - Square Feet
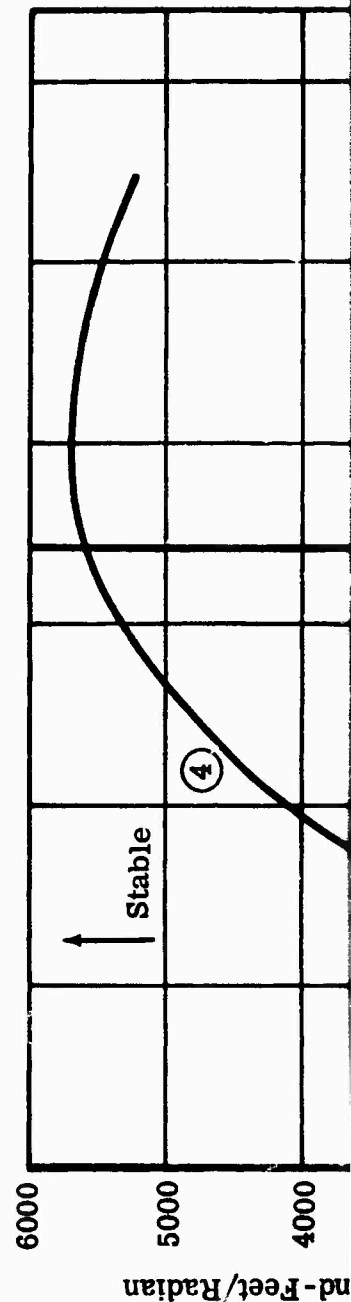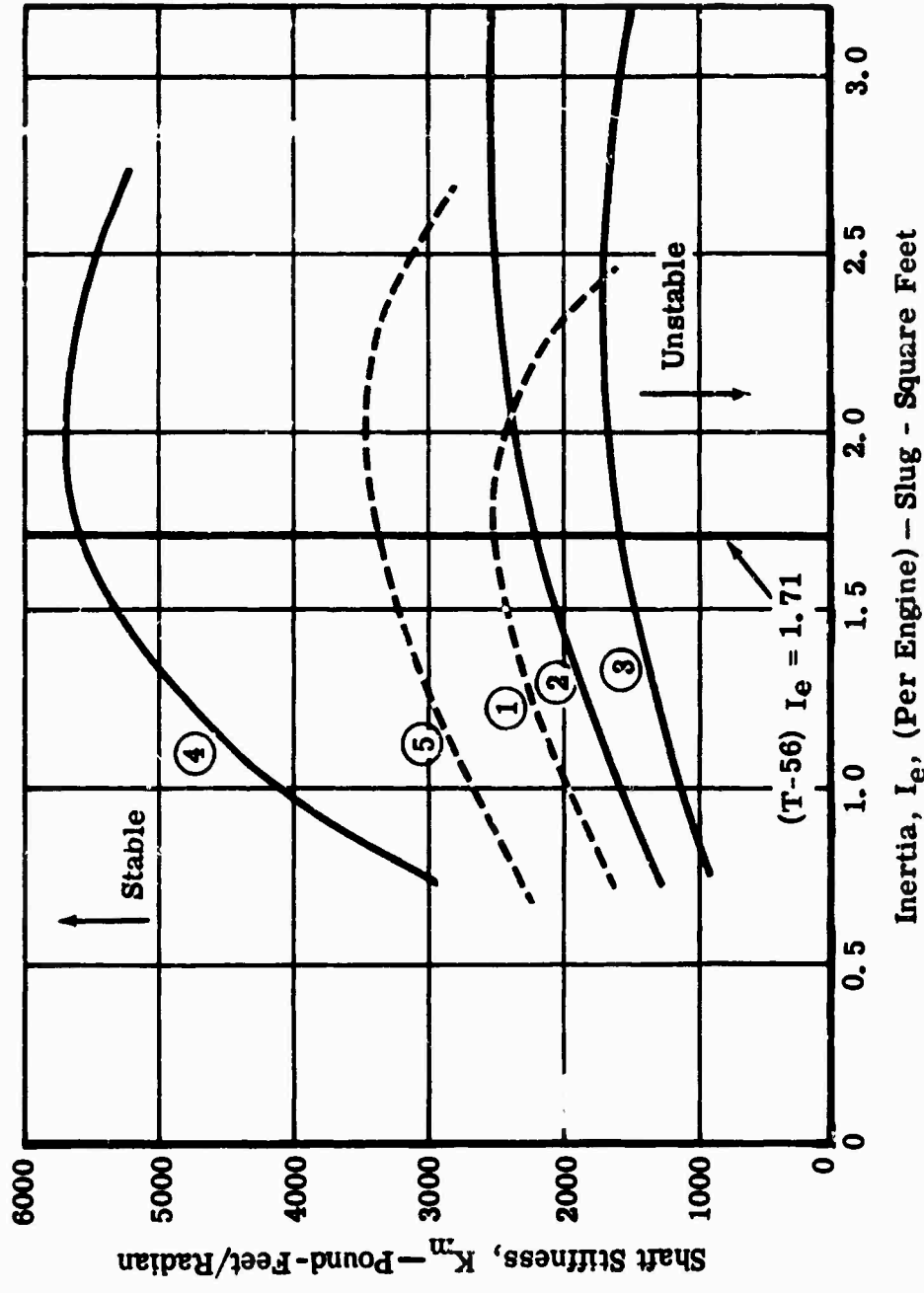
Figure 75.   HLH Torsional Stability Analog Computer Data—Four
Fixed-Turbine Engines With Regenerator

Note

N_1 – Gas Producer Speed, r.p.m.
N_s – Shaft Speed, r.p.m.
ΔN – Speed Error, r.p.m.
P.T. – Power Turbine
G.P. – Gas Producer
C.D.P. – Compressor Discharge Pressure
P_c – Control Sensing Pressure
Q_s – Shaft Torque, lb.-ft.
W_f – Fuel Flow, lb./hr.
τ – Time Constant, sec.
S – Laplace Operator
K – Governor Gain

G. P. Turbine
And Compressor

C.D.P.
Feedback

$\frac{\partial P_c}{\partial N_1}$

$\frac{\partial P_c}{\partial W_f}$

ΔN_s Error

ΔN_1 Set

$\angle N_1$ Error

ΔN_s Set

P. T. Governor

G. P. Governor

ΔW_f

G. P. Turbine

ΔN_1
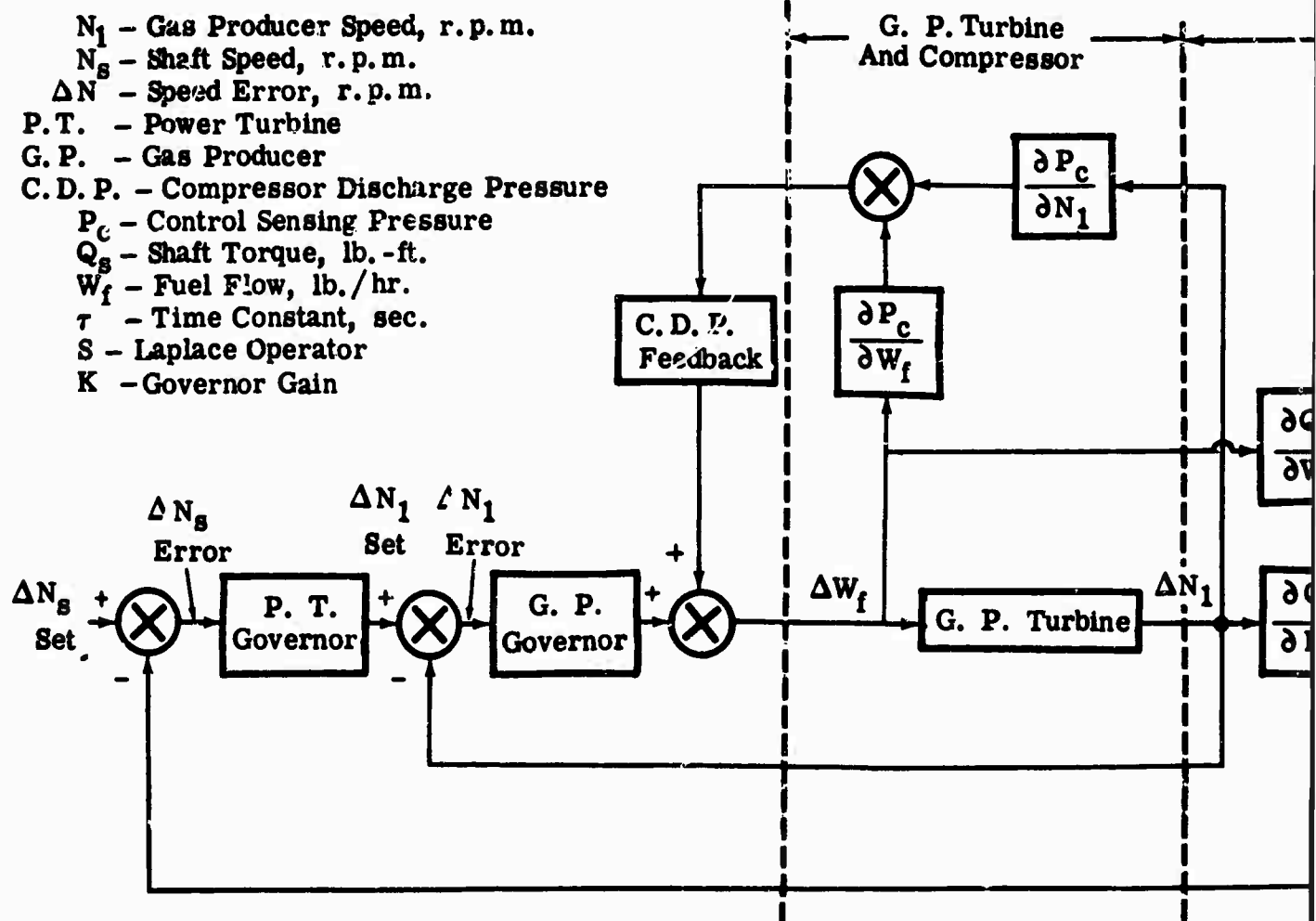
Transfer Functions:

P. T. Governor $= \dfrac{K_p}{\tau_1 S + 1}$

G. P. Governor $\dfrac{K_G}{(\tau_2 S + 1)(\tau_3 S + 1)}$

C.D.P. Feedback $= \dfrac{K_{CDP}}{(\tau_5 S + 1)(\tau_6 S}$

Gas Producer Turbine $=$

Figure 76. Model 548-C2 Engine and Fuel Control Block D

Model 548-C2 Free Turbine Engines (Cluster Of Three)

Rotor System (Rotors, Shafts, Transmissions)

G. P. Turbine And Compressor

$\dfrac{\partial P_c}{\partial N_1}$

$\dfrac{\partial P_c}{\partial W_f}$

C. D. P. Feedback

$\dfrac{\partial Q_s}{\partial W_f}$

ΔW_f

G. P. Turbine

ΔN_1

$\dfrac{\partial Q_s}{\partial N_1}$

ΔQ_s

$\dfrac{\partial Q_s}{\partial N_s}$

Rotor System

ΔN_s
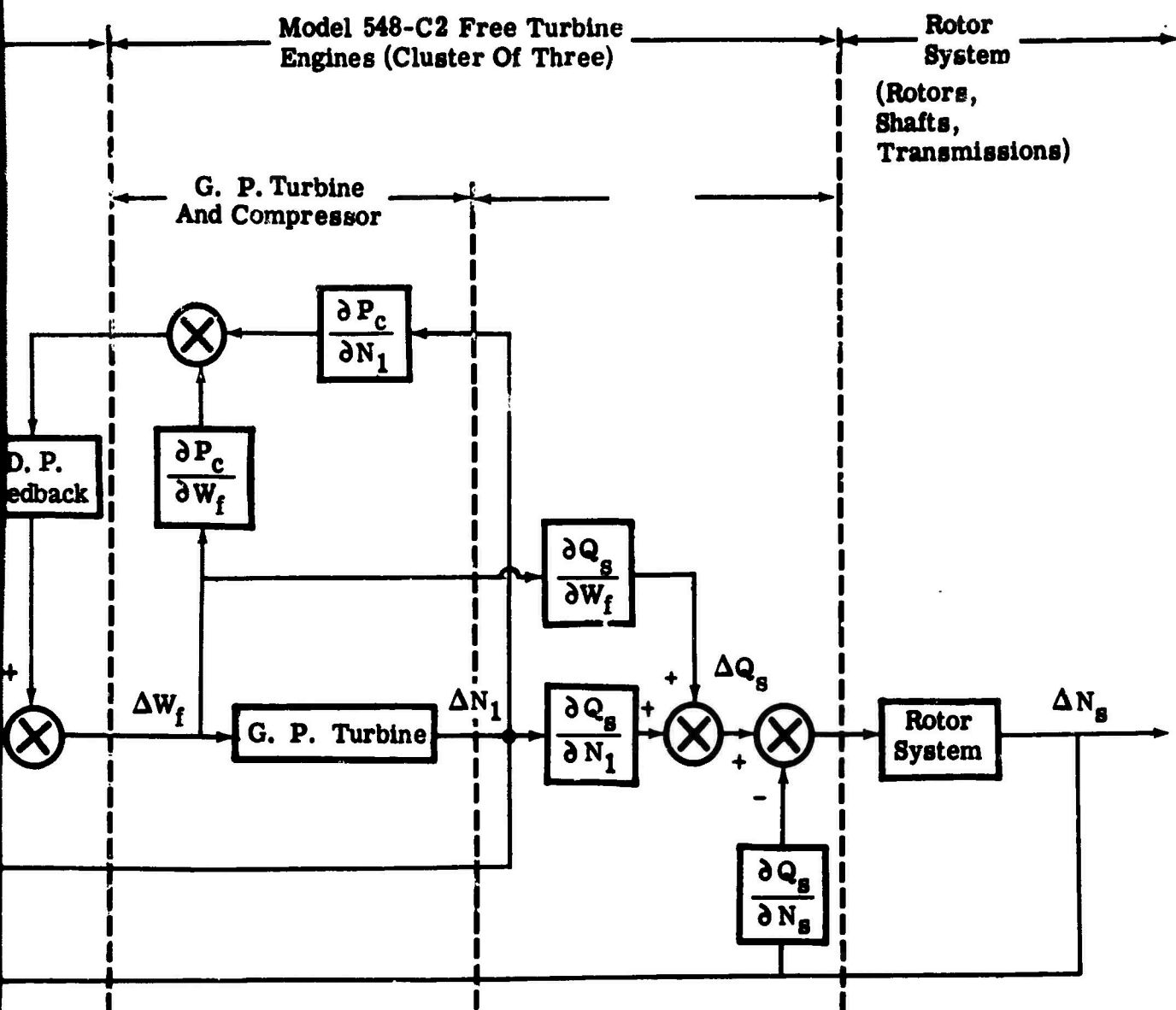
$$\text{C. D. P. Feedback} = \frac{K_{CDP}(\tau_4 S + 1)}{(\tau_5 S + 1)(\tau_6 S + 1)(\tau_7 S + 1)}$$

$$\text{Gas Producer Turbine} = \frac{\dfrac{\partial N_1}{\partial W_f}}{\tau_E S + 1}$$

Model 548-C2 Engine and Fuel Control Block Diagram

$$\frac{\partial P_c}{\partial W_f} = 0.0208 \text{ p.s.i. per pound per hour}$$

$$\frac{\partial P_c}{\partial N_1} = 0.0089 \text{ p.s.i. per r.p.m.}$$

$$\frac{\partial N_1}{\partial W_f} = 4.46 \text{ r.p.m. per pound per hour}$$

$$\frac{\partial Q_s}{\partial W_f} = 0.400 \text{ pound-feet per pound per hour}$$

$$\frac{\partial Q_s}{\partial N_1} = 0.092 \text{ pound-feet per gas producer r.p.m.}$$

$$\frac{\partial Q_s}{\partial N_s} = -0.0836 \text{ pound feet per shaft r.p.m.}$$

$$\tau_E = 0.38 \text{ second}$$

Figure 77 is a schematic diagram of a dual rotor system including forward and aft gearbox inertias and three separate power turbines and output shafts. This diagram may be easily modified to represent the single-rotor configuration—with or without drag hinge. This system is simulated on the analog computer by solving the equations of motion. The constants which appear in these equations and the nominal values used in the simulation are as indicated in the tabulation (all referred to 19,320-r.p.r. shaft) in Figure 77.

The results of these tests may be summarized as follows.

1. Single rotor configuration is torsionally stable unless, with all other values at nominal,

    a. Mast stiffness (k_m) is reduced to one-eleventh nominal value; or

    b. Power turbine inertias (I_{pt}) are increased by a factor of 4 times nominal value and mast stiffness (k_m) is reduced to one-fourth nominal value; or

    c. Gas producer governor gain (K_G) is increased to 2.5 times nominal; or

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d. Power turbine governor gain (K_p) is increased to 3.8 times nominal (droop decreased to 1.03 percent) or 3.7 times nominal with drag hinge locked, indicating that the drag hinge has a slight stabilizing effect, but does not significantly affect torsional stability.

2. Computer output traces showing this single-rotor, three-turbine performance are shown in Figure 78. The disturbance applied to the system consists of a step increase in rotor load torque, as from wind gust, causing all speeds to drop slightly. At nominal settings, the torsional natural frequency of four cycles per second is superimposed upon the natural frequency of the gas producer speed control loop, which is one cycle per second. Both oscillations are damped out after several cycles. When mast stiffness is reduced to one-eleventh of the nominal value, the torsional natural frequency is reduced to 2.5 cycles per second and oscillations become divergent.

3. Increasing the number of engines in the cluster from three to four (33-percent increase) requires a 43-percent increase in shaft stiffness for equivalent torsional stability. A 33-percent increase in power output from each of the three engines (as in a growth version) requires a 27-percent increase in shaft stiffness. The required increase is less in the second case because there is no accompanying increase in engine inertia. This would apply to fixed- as well as free-turbine engines.

4. The computer output traces in Figure 78 are for a rigid rotor system, whereas Figure 79 reveals the effect of adding drag hinges. $\Delta\xi$ is the angle of deflection of the blade about the hinge. The sudden application of a rotor load torque causes this angle to increase slightly. The number of oscillations of $\Delta\xi$ depends on the drag hinge damper coefficient. Notice that torsional stability (the oscillations of engine shaft speed, ΔN_s) is not appreciably affected by the addition of the drag hinge.

5. The dual rotor configuration is torsionally stable unless, with all other values at nominal,

    a. Engine shaft stiffness (k_s), drive shaft stiffness (k_d), aft mast stiffness (k_{ma}), and forward mast stiffness (k_{mf}) are all reduced to one-sixth nominal value; or

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Aft   Rotor
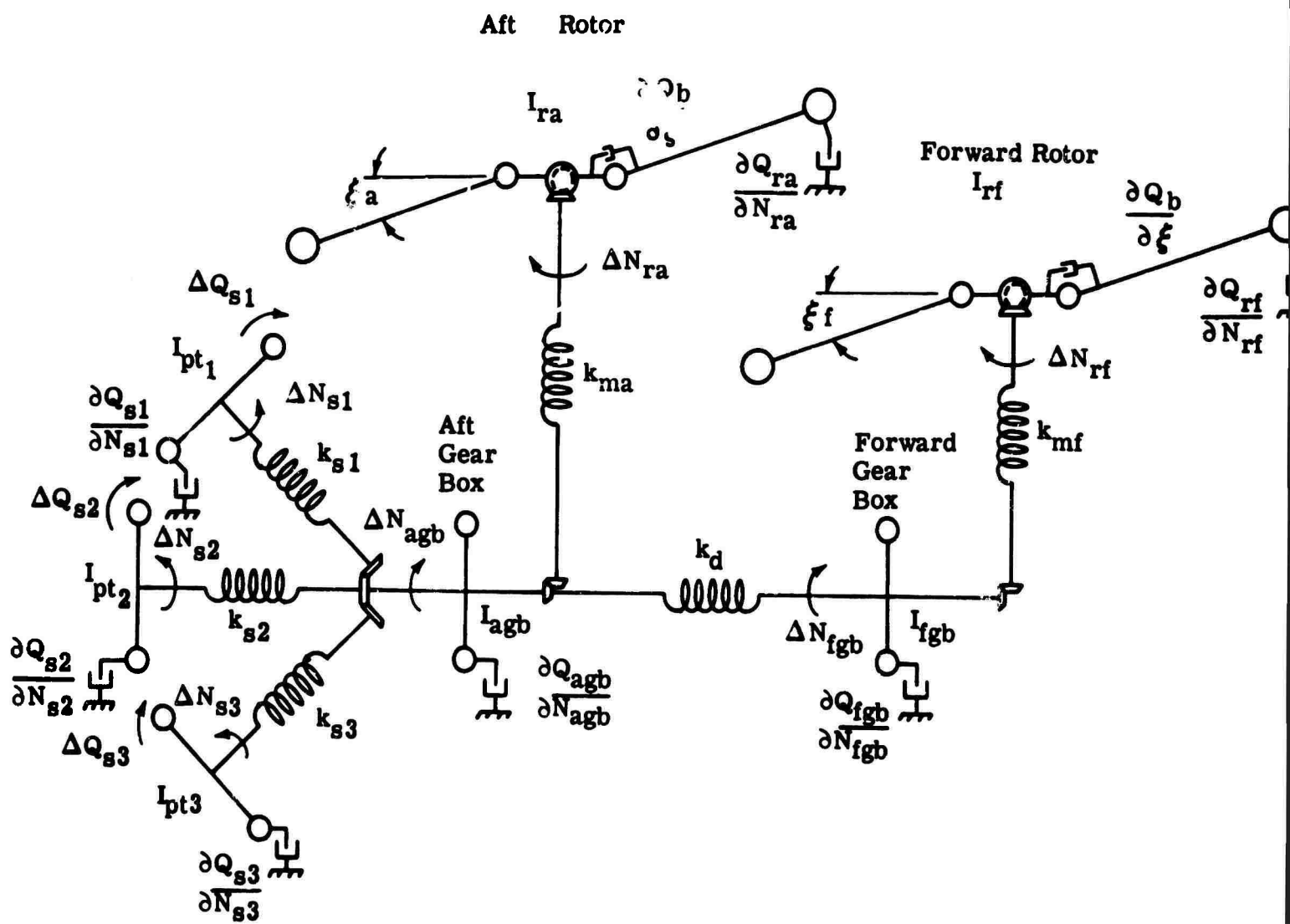
Figure 77. Complete Tan

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I_{pt} (power turbine inertia) = 0.230 slug-foot squared

$\dfrac{\partial Q_s}{\partial N_s}$ (power turbine damping) = -0.0836 pound-foot per r.p.m.

k_s (engine shaft stiffness) = 16,000 pound-feet per r.p.m.

I_{agb} (aft gearbox inertia) = 0.212 slug-foot squared

I_{fgb} (forward gearbox inertia) = 0.181 slug-foot squared

$\dfrac{\partial Q_{agb}}{\partial N_{agb}} = \dfrac{\partial Q_{fgb}}{\partial N_{fgb}}$ (aft and forward gearbox damping) = -0.1 pound-foot per r.p.m.

Single rotor with drag hinge:	19,320 r.p.m.	150 r.p.m.
I_r (total rotor inertia)	7.83 slug-feet squared	130,000 slug-feet square
bI_b (blade inertia)	7.36 slug-feet squared	122,000 slug-feet square
$b\dfrac{\partial Q_b}{\partial \xi}$ (drag hinge damper coefficient)	-500 pound-feet per r.p.m.	-8,290,000 pound-feet p
bK_b (centrifugal spring constant)	10,000 pound-feet per radian	165,800,000 pound-feet
$\dfrac{\partial Q_r}{\partial N_r}$ (rotor aerodynamic damping coefficient)	-0.50 pound-foot per r.p.m.	-8,290 pound feet per r.
k_m (mast stiffness)	2000 pound-feet per radian	32,600,000 pound-feet p
Tandem rotor:	19,320 r.p.m.	150 r.p.m.
$I_{ra} = I_{rf}$ (aft and forward rotor inertias)	3.01 slug-feet squared	50,000 slug-feet squared
$\dfrac{\partial Q_r}{\partial N_r}$ (rotor aerodynamic damping coefficient)	-0.50 pound-foot per r.p.m.	-8,290 pound-feet per r.
k_{ma} (aft mast stiffness)	400 pound-feet per radian	6,510,000 pound-feet pe
k_d (drive shaft stiffness)	1910 pound-feet per radian	31,100,000 pound-feet p
k_{mf} (forward mast stiffness)	800 pound-feet per radian	13,020,000 pound-feet p

dem-Rotor System Schematic Diagram

rtia) = 0.230 slug-foot squared

mping) = -0.0836 pound-foot per r.p.m.

ss) = 16,000 pound-feet per r.p.m.

ia) = 0.212 slug-foot squared

inertia) = 0.181 slug-foot squared

d forward gearbox damping) = -0.1 pound-foot per r.p.m.

hinge:	19,320 r.p.m.	150 r.p.m.
	7.83 slug-feet squared	130,000 slug-feet squared
	7.36 slug-feet squared	122,000 slug-feet squared
mper coefficient)	-500 pound-feet per r.p.m.	-8,290,000 pound-feet per r.p.m.
g constant)	10,000 pound-feet per radian	165,800,000 pound-feet per radian
nic damping	-0.50 pound-foot per r.p.m.	-8,290 pound-feet per r.p.m.
	2000 pound-feet per radian	32,600,000 pound-feet per radian
	19,320 r.p.m.	150 r.p.m.
rd rotor	3.01 slug-feet squared	50,000 slug-feet squared
nic damping	-0.50 pound-foot per r.p.m.	-8,290 pound-feet per r.p.m.
s)	400 pound-feet per radian	6,510,000 pound-feet per radian
ss)	1910 pound-feet per radian	31,100,000 pound-feet per radian
ffness)	800 pound-feet per radian	13,020,000 pound-feet per radian

Diagram

3
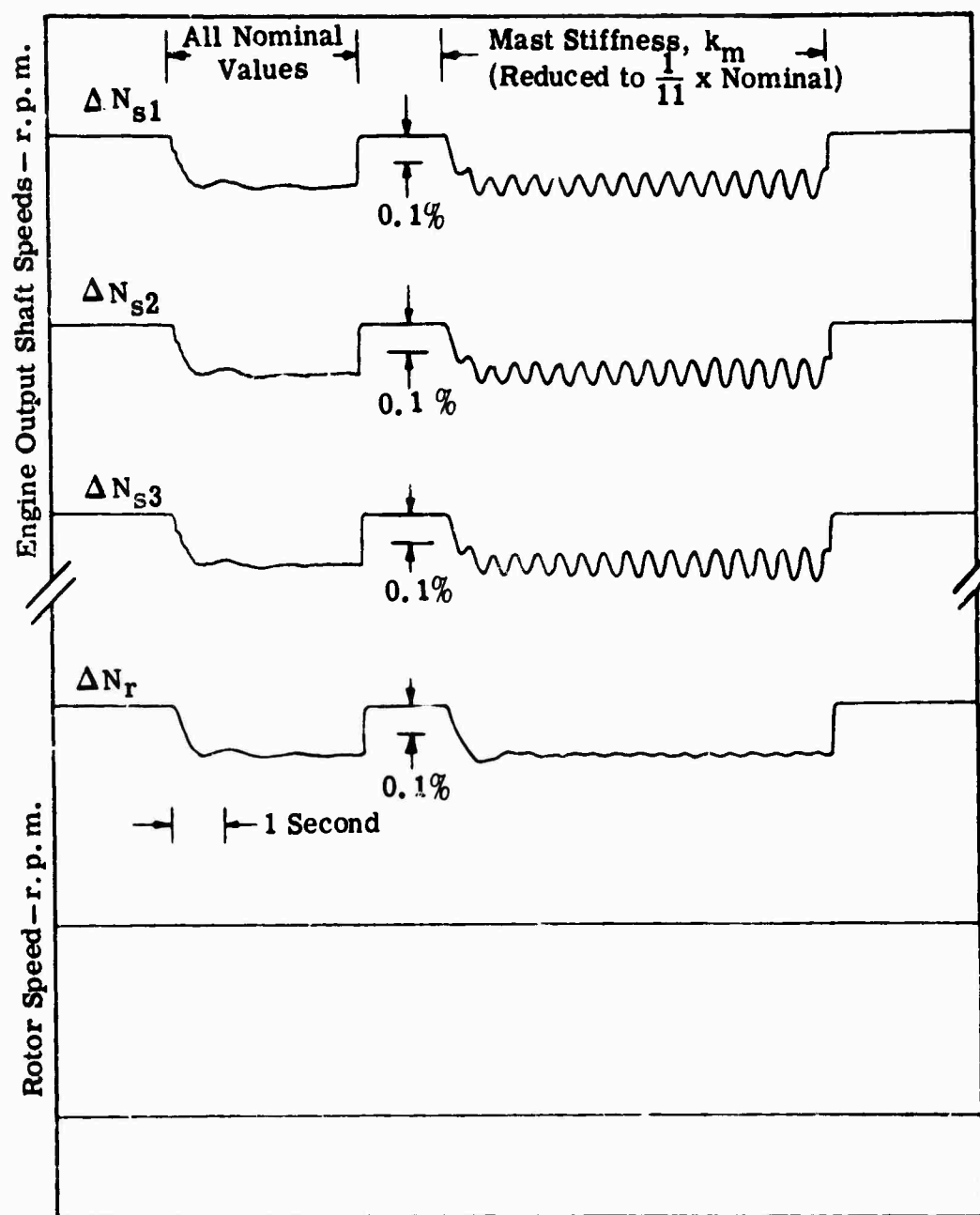
Figure 78. Computer Output Traces for Three Free-Turbine Engines,
Single-Rotor Without Drag Hinge

121
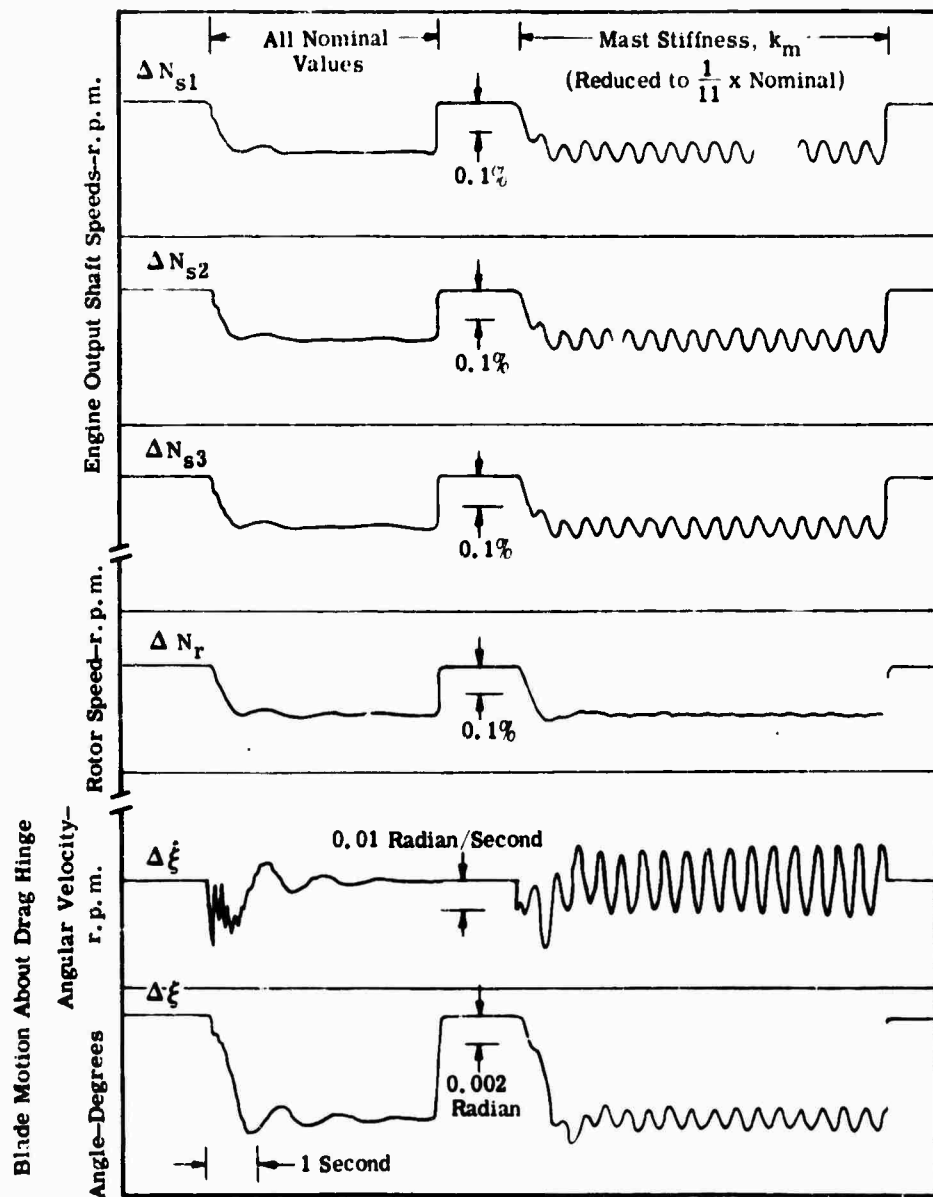
Figure 79. Computer Output Traces for Three Free-Turbine Engines,
Single-Rotor With Drag Hinge

b. I_{pt} is doubled and k_s, k_d, k_{ma}, and k_{mf} are all reduced to one-fourth nominal; or

c. K_G is increased to 2.5 times nominal; or

d. K_p is increased to 2.75 times nominal (droop decreased to 14.5 percent).

6. Figures 80 and 81 show computer output traces describing tandem rotor performance. Figure 80 shows the nominal setting performance and the effect of reducing all stiffnesses by one-sixth. Figure 81 shows the effect of a sudden 320-pounds-per-hour decrease in fuel flow to one engine. The high-frequency transient oscillations reveal the natural frequency of the power turbines and engine shafts, whereas the lower frequency is that of the power turbines and rotor system.

7. All of the traces reveal a slight tendency to oscillate at the natural frequency of the gas producer speed control loop, approximately one cycle per second. This is in the category of low-frequency stability and is to be differentiated from torsional stability. Generally it is observed in free turbine engines only. Low-frequency stability is influenced by gas producer governor gain (K_G), compressor discharge pressure feed-back gain (K_{CDP}), and gas producer turbine time constant (τ_E). This study included a limited investigation of low-frequency stability; the traces show good stability. At one point in the tests, it was found that increasing K_{CDP} by a factor of 2.5 would result in low-frequency instability.

8. A brief investigation was made of the effect of using unequal power turbine governor gains (K_p) on each of the three engines in the cluster, since some mismatch would undoubtedly exist if the individual governor concept were used. The possibility of some peculiar interaction effect was investigated but none was observed. The only effect of changing one governor gain was its effect on the net gain of the complete cluster.

POWER AUGMENTATION CONTROL

Water-alcohol injection could be employed under hot ambient temperature conditions to augment engine power. The conventional approach utilized on fixed-wing applications employs a water-alcohol system in

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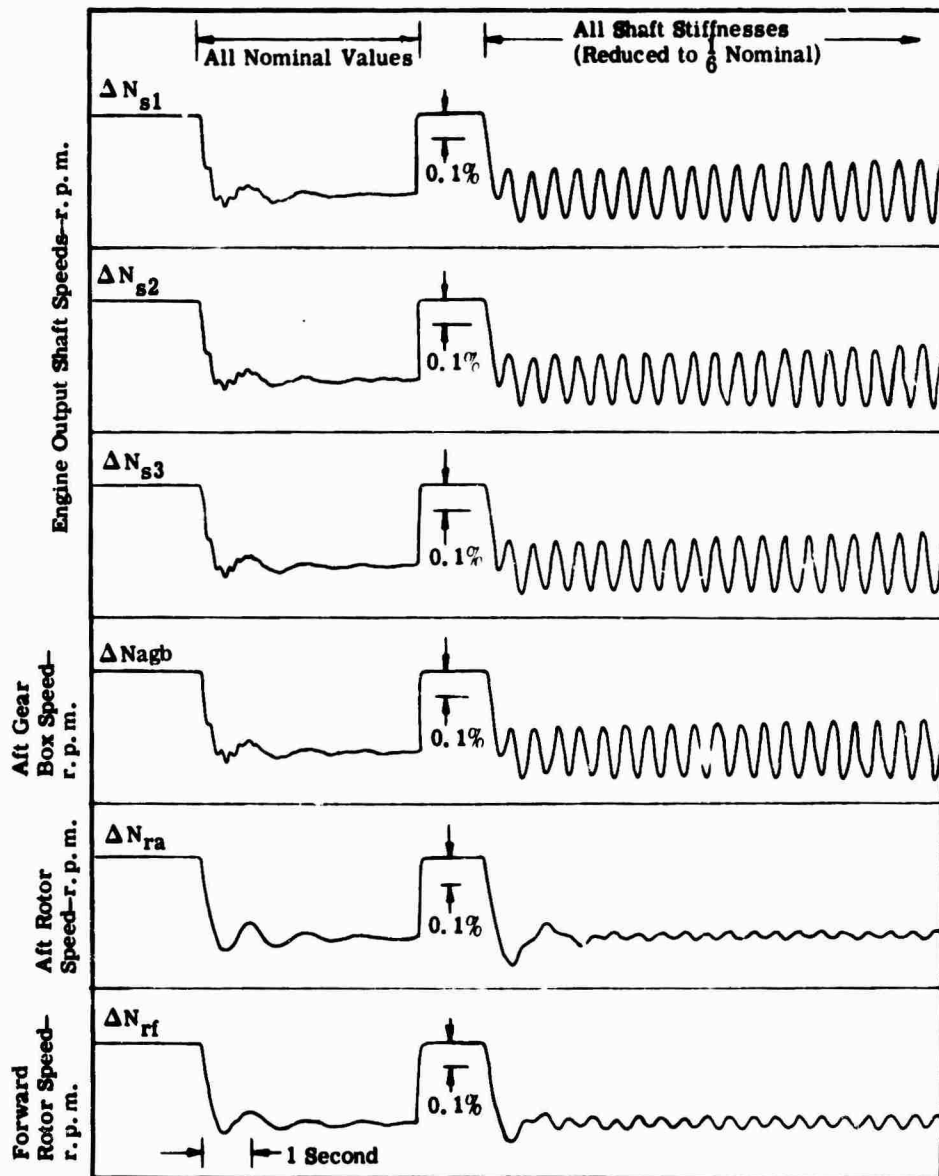
Figure 80. Computer Output Traces for Three Free-Turbine Engines, Tandem-Rotor System

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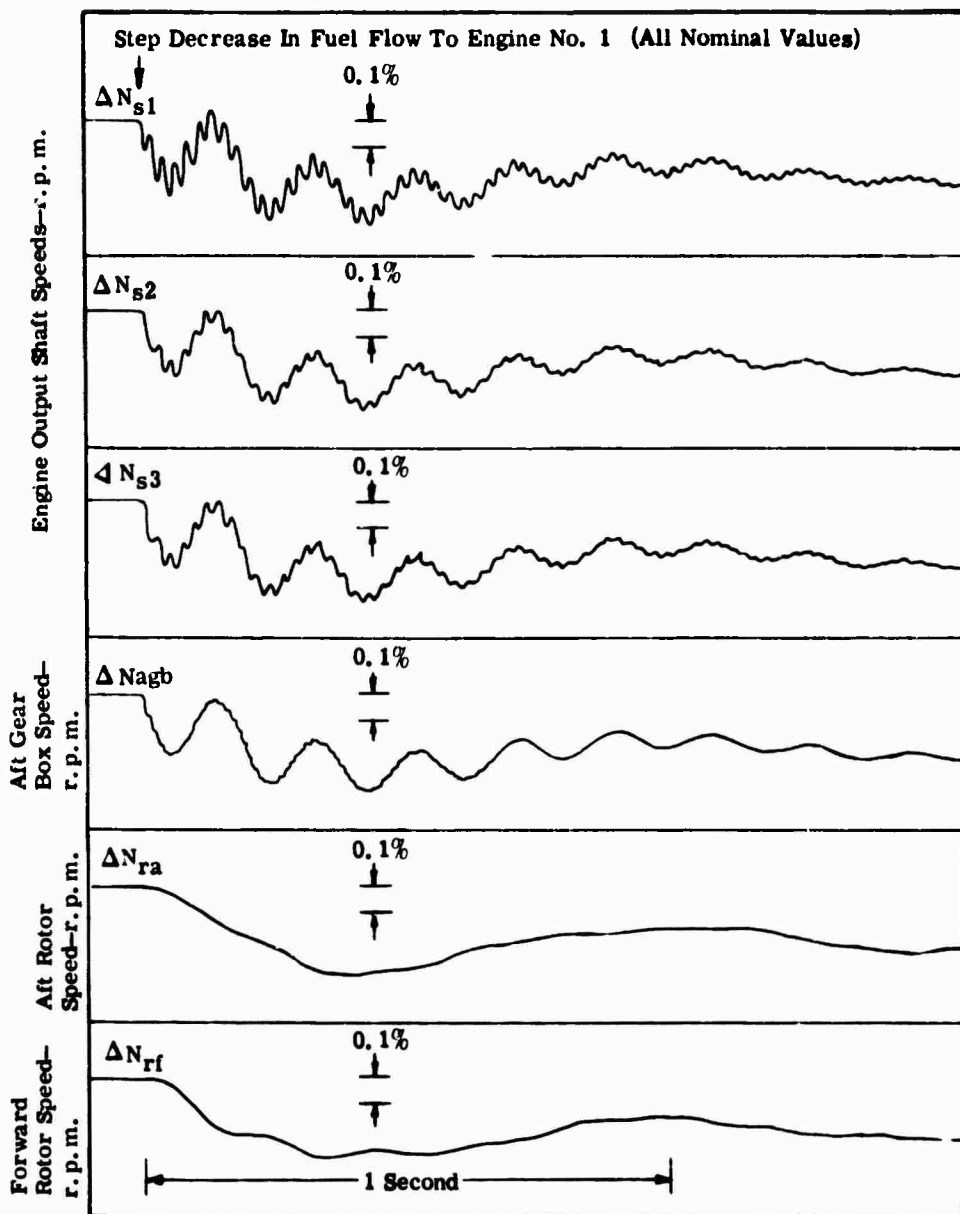
Figure 81. Computer Output Traces for Three Free-Turbine Engines, Tandem-Rotor System

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which the flow rate is not modulated (although flow is regulated at a constant flow level). All power modulation during augmentation is accomplished by regulating the turbine temperature through control of the fuel flow by power lever. This concept would also be applicable to a helicopter system in which the fuel flow is regulated by a rotor speed governor.

The selection of power augmentation would result in a boost in the available power at all turbine temperatures. The speed governor would then reduce the fuel flow as required to satisfy the load—this also means that an additional load may be applied. Associated with the water-alcohol operation would be a somewhat higher rotor speed than when nonaugmented (Figure 82), with the rotor speed increase providing the fuel reduction by the droop governor. The fixed-turbine engine would be capable of this type of operation while employing the conventional fuel control scheme and schedules. The major effect would be a somewhat higher rotor speed with augmentation; this could be eliminated by employing an isochronous governor or through the use of a "beep" system.

The free-turbine engine would require gas producer speed variations when modulating power. The use of engine inlet water-alcohol injection may dictate a somewhat different transient fuel schedule because of its effect on compressor surge and burner blowout. This is an area requiring more investigation, including testing, if such an approach appears warranted.

Selection of power augmentation in the helicopter could be either manual or automatic. In the manual selection scheme the pilot would initiate the water-alcohol flow by actuation of an on-off selection switch, and it would continue until either manually deactivated or the available mixture was depleted. Another approach would be to provide a console switch to arm the system, with an "augmentation" position on the "twist grip" to actuate the augmentation system. This would enable the pilot to arm the water-alcohol system for all critical operation, but only call on it when the nonaugmented power was adjudged to be insufficient for the load.

An automatic actuation system could be employed that would remove the need for a power augmentation decision from the pilot, but would require a more complicated control system. The augmentation could be initiated, for example, by a signal generated when the electronic temperature limiter begins controlling, providing the arming switch is on. An inadequacy of this approach is the result of the step power boost that occurs at a turbine temperature with the unmodulated augmentation

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system (Figure 83). The reduced turbine temperature, for power modulation between maximum augmented and maximum nonaugmented power, would generate a signal to deactivate the water-alcohol system, resulting in on-off cycling of the system when attempting to operate in this power region. This inadequacy could be overcome by employing the turbine temperature limiter signal to initiate the operation of augmentation, but providing a "hold-in" mechanism to continue operation of the augmentation system once initiated. The pilot arming switch would allow manual deactivation of the system when desired.

If the step power change on actuation of the augmentation system is not acceptable in a helicopter, a modulated augmentation system could be provided. The result would be a more complicated control than is required with the unmodulated approach.

OPERATIONAL COMPARISON OF FIXED- AND FREE-TURBINE ENGINES

A comparison was made of the ch aracteristics of the fixed-turbine and free-turbine engines during the various operating conditions as they apply to the HLH. The conditions considered were starting, locked rotor, ground idle, autorotation, practice autorotation recovery, speed and power stability, and rotor speed modulation.
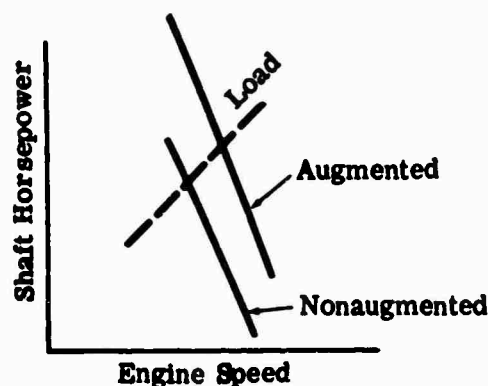
Figure 82. Governor Shift
With Augmentation
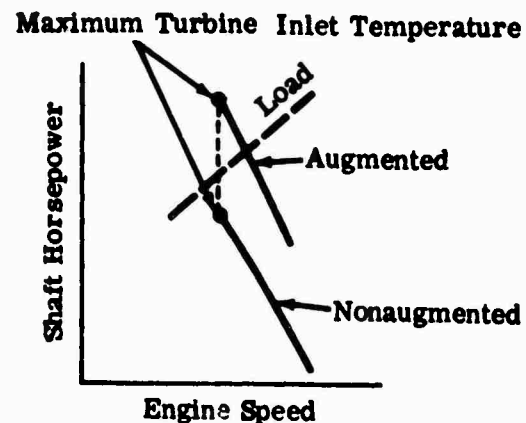
Figure 83. Power Shift
With Augmentation

Starting

The fixed turbine would require a disconnect clutch to uncouple the helicopter rotor system from the engine(s) during starting. Without this, the complete rotor load (windage and inertia) would be imposed on the gas turbine starter during engine starting—this would require a starter as large as 700 horsepower to complete a 30-second start.

With the clutch disengaged, the engine(s) could be started readily with a conventional starter of nominal power. After the completion of the start, the clutch(es) must be engaged to bring the rotor up to the speed of the engine.

The free turbine requires no special clutch because the gas generator is aerodynamically coupled to the rotor and the power turbine and rotor speeds have very little effect on gas generator operation. Thus, the gas generator(s) can be started with a conventional starter (on the order of 100 horsepower).

Locked Rotor Operation

The capability of checking the operation of the engines without the rotor revolving may be desired on a multiengine installation. With the clutches disconnected, the fixed-turbine engines can be operated at zero power for engine and control-system checkout, without rotor system rotation. No special locking mechanisms would be required.

When employing free-turbine engines, a rotor brake mechanism is required to allow engine operation without the rotor system turning. This brake would allow operation at ground idle power for engine and control system checkout.

Ground Idle

At ground idle, minimum fuel consumption is desired. Reduced rotor speed operation is desired to reduce rotor windage at minimum collective pitch. For low-speed ground idle operation with a fixed turbine, the engine and rotor must be maintained at a speed of 80 percent or greater (Figure 84) when considering operation at 130°F. ambient. Also, caution must be employed to ensure that the collective pitch (load) is not increased until the rotor and engine speed transient (from ground idle to normal operational speed) has been nearly completed. Early application

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of the collective pitch could result in total "bog-down" of the rotor and engine because the load becomes greater than the engine power available.

The ground idle rotor speed with a free turbine can be as low as 50 or 60 percent. Because the rotor (load) is not solidly connected to the gas produce r, an increase in the collective pitch (load) would not affect acceleration of the gas producer. The increasing engine torque due to the gas producer speed increase would arrest any rotor speed bog-down resulting from the application of collective pitch while at low (ground idle) rotor speeds.

Autorotation (Decoupled)

During the autorotation maneuver, the rotor system must be decoupled and the gas turbine engine must operate at zero power.
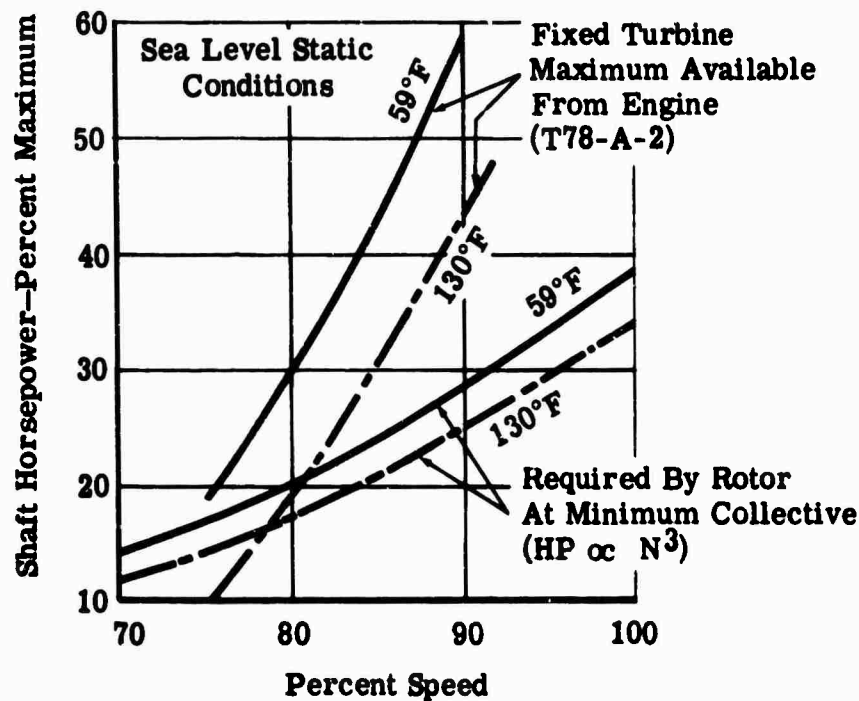
Figure 84. Fixed-Turbine Characteristics at Low Power

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Decoupled operation with a fixed turbine engine would impose no control problems. Decoupled (zero horsepower) stability would be achieved with a proportional speed governor of 3- to 5-percent droop.

The low frequency stability of a free turbine during decoupled operation would not normally be as good as that with a fixed-turbine engine. This is because of the additional lag resulting from the inertia of the gas producer rotor, in conjunction with the very low inertia of the power turbine when not coupled to the rotor system. This may require precaution and/or compromise in the design of the governor system, or the acceptance of the low-amplitude, low-frequency-limit cycle oscillations during autorotation operation.

Practice Autorotation Recovery

Practice autorotation is a conventional maneuver for helicopters and is assumed to be applicable to the HLH. This maneuver involves the simulation of a total loss of engine power by reducing the engines to idle. The engines are kept running at minimum power during the practice maneuver to permit the rapid increase in power needed for recovery in the event of pilot error during practice landing.

If the power recovery is attempted with the fixed turbine engine at (or near) 100-percent speed, the power buildup would be very rapid, being limited only by the rate at which the fuel flow (and compressor variable geometry) can be increased. However, a rapid recovery from a reduced rotor (and engine) speed would not be possible. This is because the power available at engine speeds less than 95 percent would not be sufficient to support the rotor pitch (load) that would be selected in attempting the recovery (Figure 85).

The response of the free turbine during a powered recovery would be dependent on the acceleration characteristics of the gas producer. The power response would not be as fast as that with a fixed turbine, but would lag the load because of the inertia of the gas producer rotor, coupled with gas turbine operating limits of compressor surge and turbine gas temperature. A reasonable elapsed time from zero to maximum power, for an engine in the 4000-horsepower class, would be 3 to 5 seconds.

Since the free turbine engine gas producer rotor is aerodynamically coupled to the power turbine and load, the power turbine speed essentially does not affect the acceleration characteristics of the gas producer. Thus, a power recovery from a reduced rotor speed condition

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**Free Turbine Power Available At
Maximum Gas Producer Speed (548-C2)**
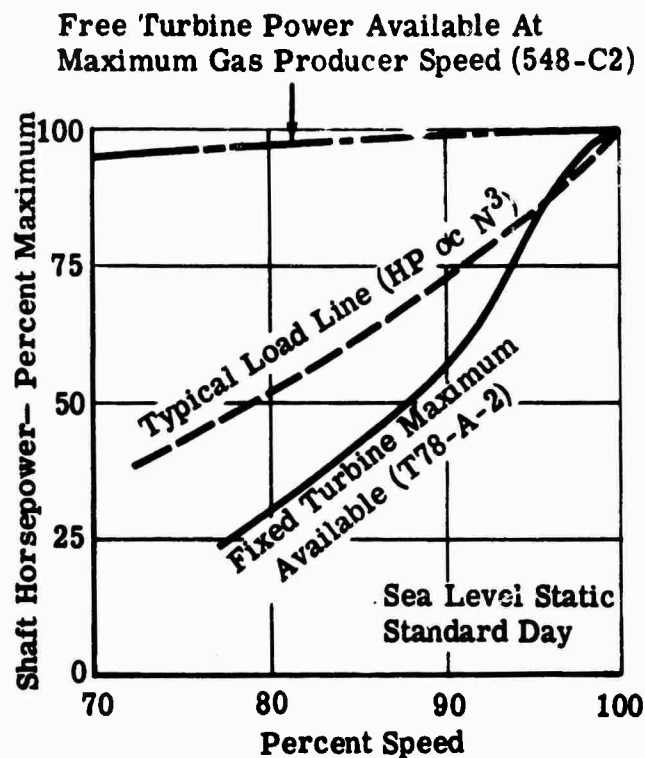
Figure 85. Maximum Power Versus Speed
for Fixed and Free Turbines

during a practice engine-off autorotation landing could satisfactorily be accomplished, assuming the 3- to 5-second power response is satisfactory.

Speed and Power Stability

In general, except for decoupled operation, the critical power level for stability is at the high powers. Initial studies have indicated that torsional oscillations will not be a problem in this power class and application. It should be recognized that in the event the rotor shafting design is such that the torsionals do occur, the high inertia of the fixed-turbine engine complicates the system stability problem. Suitable low-frequency stability should be obtainable with a proportional governor droop of 5-percent speed.

The low inertia of the power turbine results in the free-turbine engine being better than the fixed-turbine engine with regard to torsional

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stability. In general, the free-turbine engine would have less low-frequency stability margin than a fixed-turbine engine. This is because of the added lag due to the inertia of the gas producer rotor.

Rotor Speed Modulation

In the operation of the helicopter it may be desirable to vary the rotor speed to achieve optimum performance.

The fixed-turbine engine has a limited usable range of speed for unrestricted collective pitch modulation (Figure 85).

The free-turbine engine will allow a wide range of rotor speed operation. This is possible because the gas producer is not mechanically connected to the rotor, and thus operates independently.

STARTING ANALYSIS

The multiengine requirement of the HLH and the consideration of larger engines than have been used in helicopters prompted an investigation of engine starting. This investigation included consideration of various system arrangements, determination of starting torque requirements for the engines studied, and an analysis of control operation during the start cycle.

System Arrangement

The various pneumatic starting systems shown in Figure 86 were investigated for fixed- and free-turbine engines. The simplest fixed-turbine approach shown is that of Case 1. This system assumes that all engines are started simultaneously by either GSE or on-board air supply. Note the minimum number of valves and the lack of clutches. An analysis was made which showed that a 650-horsepower starter system would be required to accelerate the power train to ground idle within 30 seconds. This type of arrangement would require a large on-board air supply for air starts and ground starts in remote areas. This approach is considered inadequate for the HLH.

A system which provided sequential starting utilizing clutches and multiple starters (Case 2, Figure 86) was considered for the fixed turbine. This arrangement required the addition of clutches and associated controls and starter shutoff valves. Available starters and GSE can be used with this system. Cases 1 and 2 depend solely on the on-board air

Case 1   Case 2   B
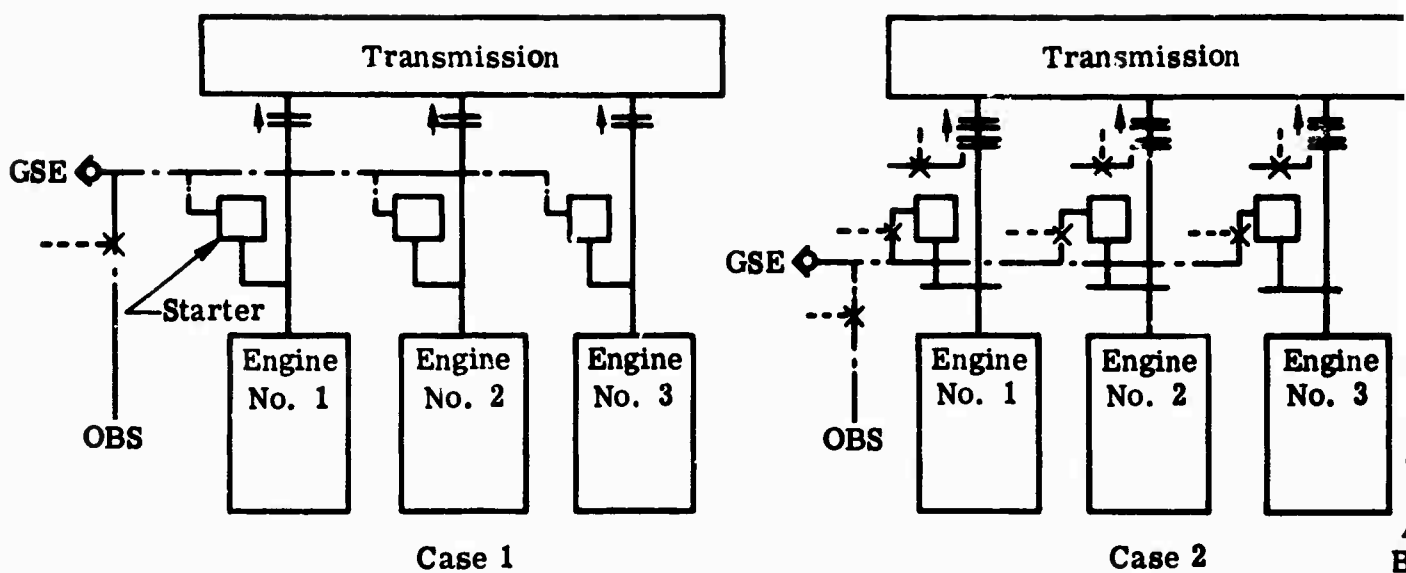
—·——   Pneumatic

------   Electrical

Overrunning Clutch

Clutch and Control

GSE   Ground Support Equipment

Shutoff Valve

Check Valve

OBS   On Board Supply

Figure 86. Start S

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Fixed Turbine

Transmission

Engine
No. 3

Engine
No. 1

Engine
No. 2

Engine
No. 3

GSE

A/C
Bleed

OBS

Case 3

Transmission

GSE

Engine
No. 1

Engine
No. 2

Engine
No. 3

OBS

Case 4

Free Turbine

Transmission

OBS

GSE

Engine
No. 1

Engine
No. 2

Engine
No. 3

A/C
bleed

Case 5
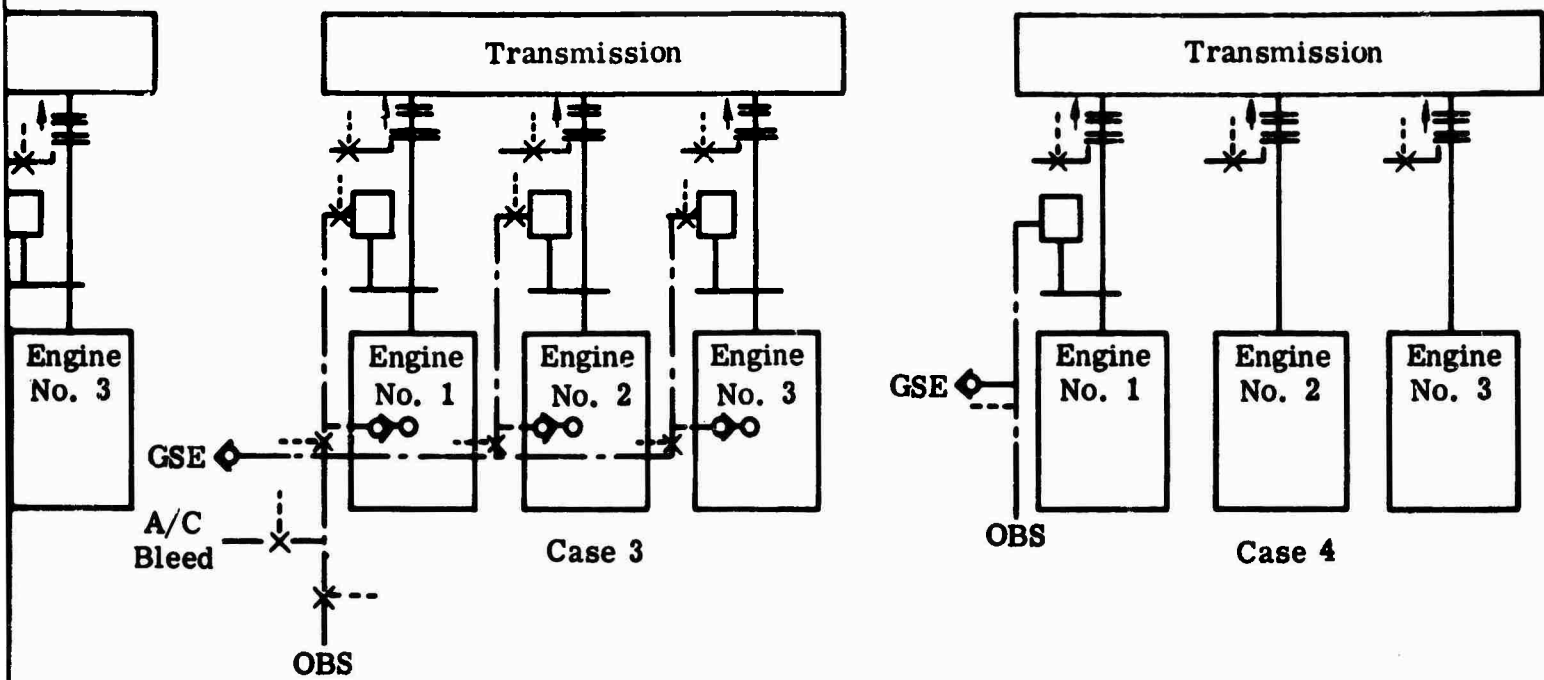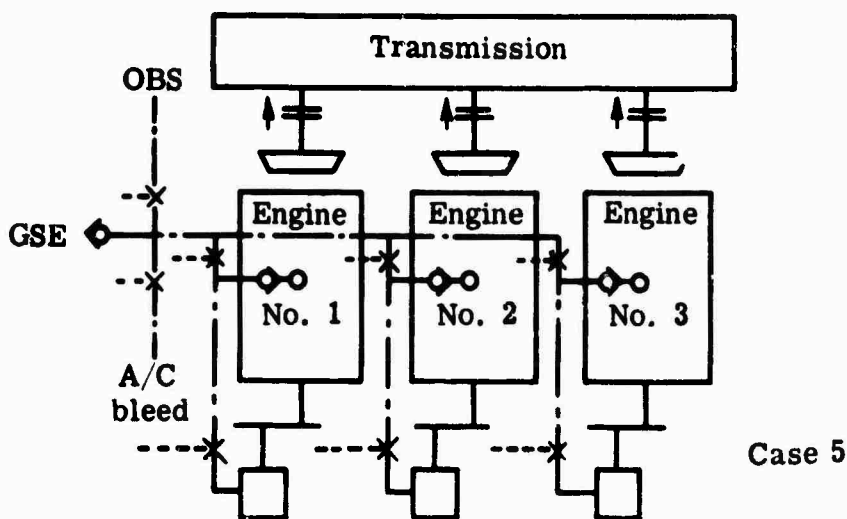
8. Start System Schematic

supply for air starts. Engine motoring by the power train is not feasible because of the overrunning clutch unless lock-out devices are employed.

Case 3 depicts a system which is very similar to the Electra and P3A installations. This arrangement provides sequential starting utilizing air bleed. Once one engine is started it can be used to start the remaining engines or to assist the GSE; this results in minimum cycle time. More important, however, is the ability to use engine bleed for air starts, greatly improving the reliability of the start system. No longer is the pilot solely dependent on the auxiliary air supply. Very little weight penalty is incurred by providing bleed air for starting, since bleed air is usually required by the aircraft for ice protection, etc.

Another system (Case 4) was considered which provided sequential starting utilizing clutches and a single starter. Number one engine would be started and accelerated to 90-percent design r.p.m. At this point, the clutch would be engaged and the transmission and rotor system would be accelerated to 90-percent design speed with the rotor blade angle set at minimum. Then the clutch would be engaged on engine two and, after starting this engine, the clutch would be engaged on engine three.

This system appears simple from an installation viewpoint; however, it is inadequate from an operational viewpoint. To use this approach, the overrunning clutch must be modified to permit lock out. The dependence on the successful start of engine number one before engines two or three could be started would reduce the operational flexibility of the HLH.

Case 5 (Figure 86) is a start system schematic for a free-turbine installation. This system is identical with the Case 3 fixed-turbine system, except that the clutches and associated controls are replaced by the free turbines. This system, which provides sequential starting utilizing air bleed, offers a high reliability for air starts (three energy sources instead of one), on-board ground start capability, and minimum starting time.

Maximum use of vehicle subsystems and low start torque requirements has led to the use of hydraulic and electrical starters in helicopters. The analysis made for the pneumatic systems would also apply to these systems if hydraulic motor/pumps or starter/generators are used. Redundancy of the starter power source, such as provided by engine bleed in Cases 3 and 5, could be provided in this manner.

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Starting Torque Requirements

The engines considered in this study are of two basic designs—T78 and T56—and as such can be categorized into two groups in regard to starting. The fact that both free- and fixed-turbine engines are within each group should not affect the basic start torque requirements. This is based on the assumption that very little or no assist is obtained from the latter stages of the fixed turbine during the critical phase of starting. However, in the determination of time to reach ground idle, the difference between mass moment of inertia of a gas producer section and a complete engine must be considered.

The torque required during motoring and that available once self-sustaining speed is obtained versus engine speed is shown for the T78 type of engines in Figure 87. Also included on this figure are the estimated effects of ambient temperature on the torque characteristics. In general, the shapes of the characteristic curves are similar except for the perturbation of the -65°F. curve in the 6000- to 9400-r.p.m. range. This results from the anticipated shift in compressor stall characteristics with temperature on an observed speed basis. This plot indicates that a starter capable of producing approximately 25 horsepower at 3400 r.p.m. (engine) would be satisfactory for the most severe condition (-65°F. ambient). Use of a starter of this capacity should result in standard-day start times of approximately 18 seconds for a fixed-turbine and approximately 12 seconds for free-turbine versions of the Model T78.

Figure 88 presents (for the T56-type engine) the required torque during motoring and that available for acceleration, once self-sustaining speed is obtained. The shape of the curves for various ambient temperatures is very similar to the T78, but the motoring requirements are somewhat higher. This results from the higher airflow capacity of the T56.

A starter with a capacity of 41 horsepower at 2400 r.p.m. would be required to start the T56-type engine on a -65°F. day. Use of a starter of this capacity should result in standard day start times of approximately 18 and 12 seconds for fixed- and free-turbine engines, respectively.

A number of starter types were investigated for suitability for the powerplants considered in this study. These were air turbine starters, air-breathing starters, hydraulic motor-pump starter systems, and electrical starter-generators. Starting requirements for the turboshaft engines

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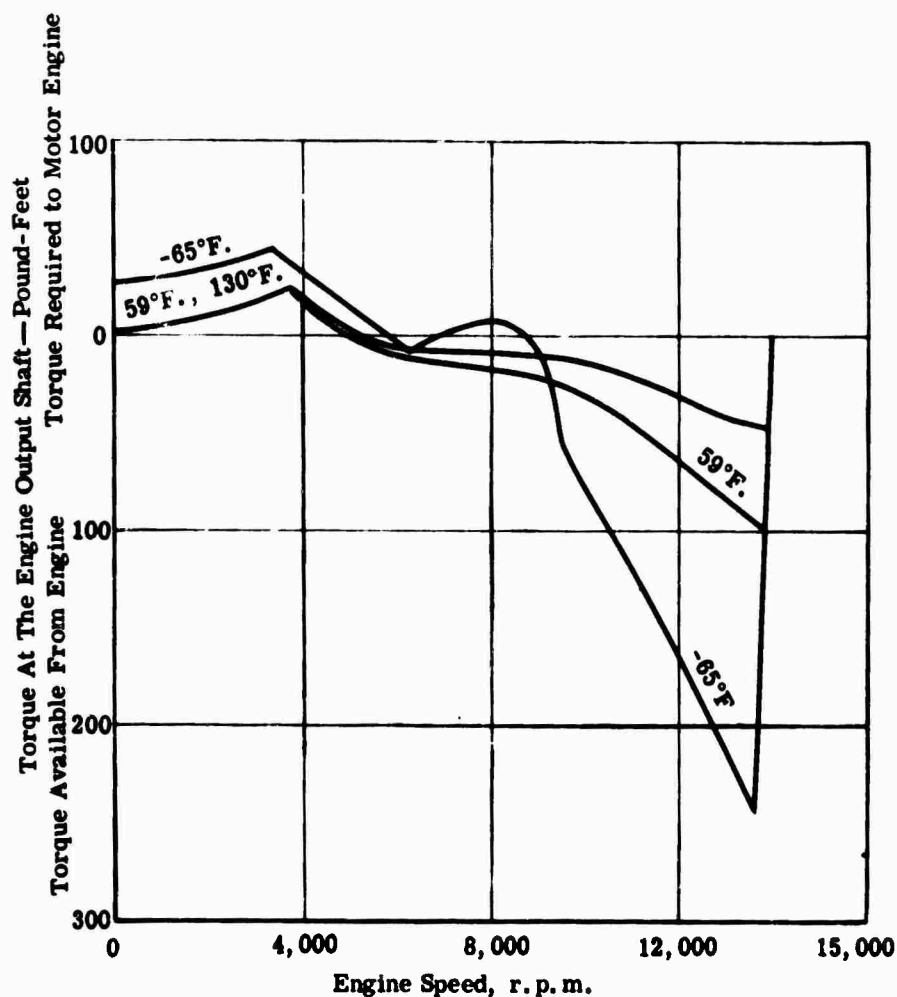
Figure 87. Estimated Starting Characteristics for Models
545-C2, 546-C2, and 548-C2

considered in this study are modest enough that any of these starting
systems would be adequate. It is recognized that current aircraft de-
sign practice is to integrate the starter system into the airframe auxiliary
power system. For this reason, a specific starter configuration has not
been recommended.

137
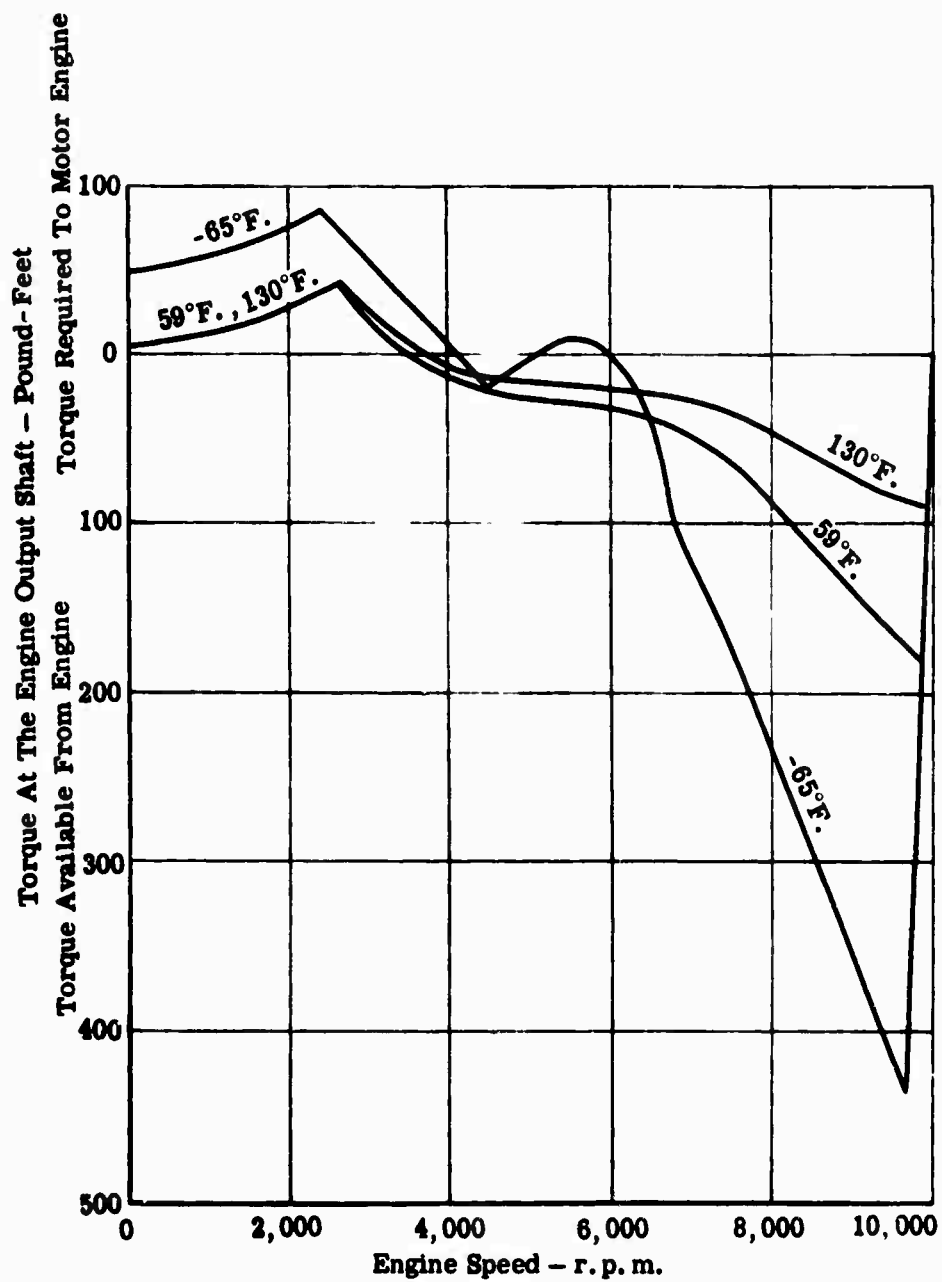
Figure 88. Estimated Starting Characteristics for Models
501-M25 and 501-M26

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Fuel Control Operation During Starting

To accomplish start-up, the starting motor cranks the engine to approximately 15-percent speed, at which time fuel flow and combustion are initiated to assist in engine acceleration. When the engine (and starter) reaches 50- to 60-percent speed, the engine will develop sufficient turbine power to complete the acceleration to operating speed without starter assist.

The controlling or scheduling of fuel flow to the engine during starting must be such that it provides the maximum turbine power consistent with the engine operating limits of compressor surge and turbine temperature. Fuel scheduling is accomplished by sensing appropriate air pressures and temperatures and the engine speed to establish the desired fuel flow. The specific parameters to be utilized are dependent on the type of engine and the steady-state scheduling and controlling employed. The clustering of several engines, and the fact that the HLH is a large vehicle, do not dictate any special scheduling considerations.

The steady-state control is to be accomplished by governing the output shaft speed by regulating the fuel flow. This closed-loop concept allows a fair degree of latitude in the selection of fuel schedule bias parameters. In order to compensate for air density changes associated with varying altitude, the scheduling action must be biased by an air pressure signal, and this could be either compressor inlet or discharge pressure. Compressor discharge pressure sensing is desired, since during starting and engine acceleration it provides corrective action in the event of a compressor surge. In addition, the icing problem that can occur with the probe located at the compressor inlet is eliminated.

Start and acceleration fuel scheduling must also be biased by an air temperature signal to obtain the correct metered fuel flow for maintaining a turbine temperature. This bias temperature could be sensed at either the compressor inlet or discharge. Compressor discharge temperature, in theory, would be the most accurate. However, it requires sensing a gas of a moderately high temperature level and large temperature range, and can impose undesirable lags during the engine speed transient due to sensor (and servo system) lag. Compressor inlet temperature is desired because of its temperature level, range, and insensitivity to engine speed.

On regenerative gas turbine engines, another air-temperature scheduling parameter is required. To compensate for dynamic lag and steady-

state variations in the regenerator, the burner inlet temperature must be sensed if reasonably precise scheduling is required during starting and acceleration. The burner inlet temperature parameter would be employed in conjunction with compressor inlet temperature.

The start fuel schedule must be established as a function of engine speed, and for a free turbine engine it is the gas producer speed. This signal allows adjustment of fuel schedule for the airflow variation associated with speed, including the effects of variable compressor geometry (vanes, bleeds, etc.). It also provides sequential signals to operate an automatic starting system and any compressor variable geometry.

A closed-loop temperature limiter is desired to protect against extreme turbine temperatures during starting. Since closed-loop turbine temperature limiting is desired during steady-state operation, it can be provided for starting protection without significantly increasing system complexity. Closed-loop turbine temperature limiting would be accomplished by thermocouple sensing the gas temperature and using an electronic amplifier control to compare the thermocouple signal to a reference in order to establish the turbine temperature error. This would then be transmitted to the fuel control to modify scheduling.

Figure 89 is a functional diagram of the start and acceleration scheduling portion of the fuel-control computer for a nonregenerative engine. This hydromechanical scheduling with closed-loop limiting is identical to the approach to be employed in the modern T56 control, and to be employed on the Model 548-C2 engine.

For a regenerative engine, the scheduling is very similar, with the addition of the burner inlet temperature compensator. This is illustrated functionally in Figure 90 and is identical to that of the T78-A-2.

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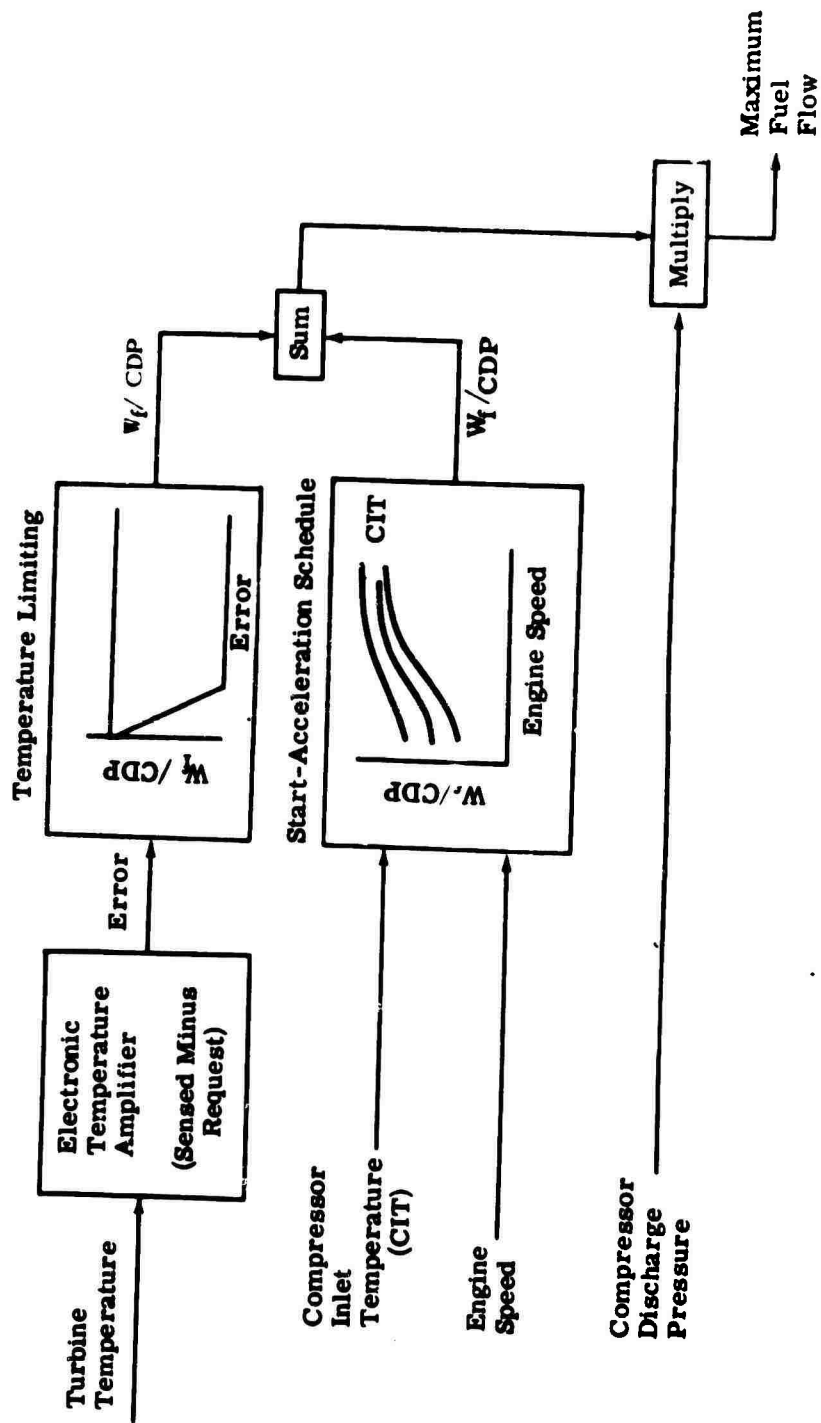
Figure 89. Start and Acceleration Schedule for a Nonregenerative Engine

141
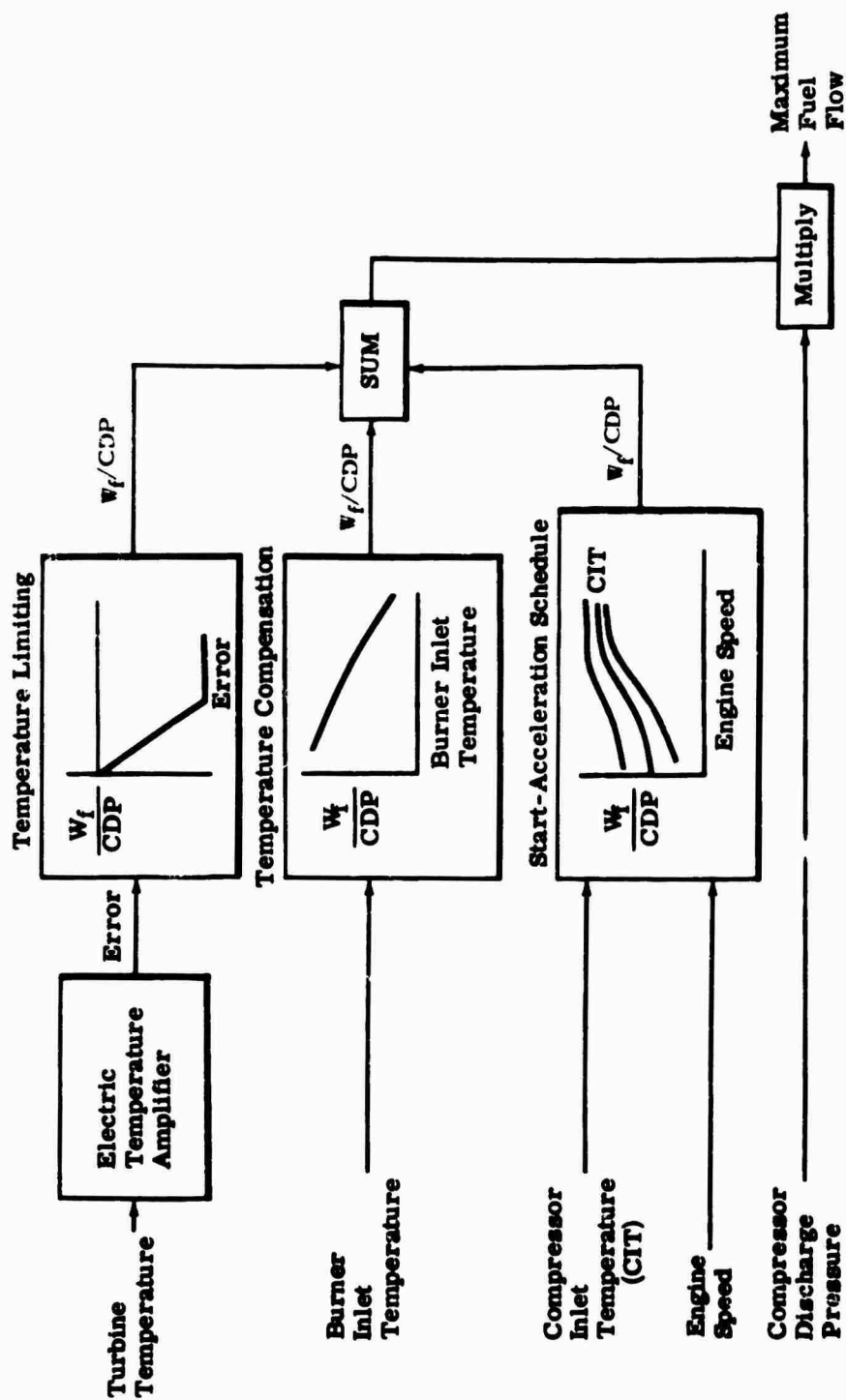
Figure 90. Start and Acceleration Schedule for a Regenerative Engine

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(U) POWER COUPLING STUDIES

The design gross weight of the HLH is such that the power requirement will be more than double that installed in the largest helicopter now in service. This large power requirement will dictate the use of a minimum of three engines and possibly four, depending on the final vehicle configuration. The problems associated with combining the power output of the engines and the transmission of this power to the rotor(s) are of vital concern to engine, transmission, and vehicle designers. In addition, the use of large diameter rotors (100 feet) and the limitations on rotor tip speed (650 to 700 feet per second) require that a large speed reduction be provided between the engine and the rotor shaft.

These problems are discussed in this section, which includes subsections on combining and reduction gearing, shafting, clutches, safety devices, and gas-coupled rotor drives.

COMBINING GEARING

Use of multiple engines in the HLH will require combining of power prior to or following entry into the main transmission. Early in the study, it was recognized that combining power prior to main transmission entry should be avoided, if possible. A simpler and lighter design results by integrating the engine combining gears into the main transmission so that redundant gear cases, seals, bearings, and gears are eliminated.

Geared transmissions were investigated which provided for engine installation parallel to the vehicl axis, and for installation skewed at an angle sufficient to provide engine clearance and still permit driving into a single spiral bevel gear with no offset.

Engines mounted parallel to the axis of the vehicle have some advantage because the inlets point straight forward for better ram air pressure recovery. However, at the low forward speed now planned for HLH, the beneficial effect on performance is at best negligible and may not exist when rotor downwash vectors and fuselage interference from a specific installation are considered. Scale effects introduced by the size of the HLH make the use of skewed engines quite attractive. Transmission design is simplified and frontal area of the propulsion package is compatible with the fuselage widths currently being considered. In this approach, the power from each engine can be directed through an individual right-angle gear mesh which also serves as part of the

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overall speed-reduction train. The gear and bearing problems associated with a right-angle drive are greatly reduced, since the power-combining function is delayed until the corner has been turned.

Two basic transmission designs were investigated in sufficient detail to obtain insight into the combining gearing problem. Because of the difficulty in separating the combining function from the total speed reduction function, complete transmissions were considered. The design parameters assumed for this study were as follows:

- Design Input Power   15,000 horsepower
- Input Speed   19,300 r. p. m.
- Rotor Speed   150 r. p. m.
- Cross-Shaft Speed   6,000 or 19,300 r. p. m.
- Maximum Continuous Moment On Rotor Shaft 1,500,000 pound-inches
- One Alternator Drive
- One Tachometer Drive
- Rotor Brake and Lock Required

Parallel Arrangement
_____

Two parallel combining gear arrangements, directly coupled to a differential planetary final reduction stage, were studied. The first arrangement (Figure 91) combines four Model 548-C2 engines into the main transmission housing. The second arrangement (Figure 92) combines four regenerative engines into a similar housing. The basic difference between the two arrangements is the distance required between the engines. The regenerative engines require a minimum distance of 36 inches between engines and the Model 548-C2 engines require a minimum distance of 24 inches between engines. For the 24-inch center distance, the engines drive double helical pinions which mesh with a common bull gear. The 36-inch center distance requires an idler gear between the pinions and the bull gear. In each case, an overrunning clutch is provided on each engine shaft for "engine-out" conditions.

The main transmission housing contains two bevel pinions driven by the combining bull gears. These pinions mesh with a common bevel ring gear to provide a single shaft input to the differential sun pinions. The sun pinion drives twelve planets which mesh with an output ring gear and two fixed ring gears. The angular spacing between the planets is maintained by the tooth reactions from the ring gears, making a planet cage unnecessary. At low speeds when centrifugal force acting on the planets does not exceed the inward tooth reaction component, the radial

Overrunning Clutch — Enlarged View

548-C2
T78 Free Turbine

19,320 r.p.m.
3,750 h.p.

548 C2
T78 Free Turbine

Symmetrical Engine Arrangement —
Two of Four Engines Shown

150 r.p.m.
5,350,000 in./lb.
12,750 h.p.

Heli
Spu
Heli

1730 r.p.m.

4225 r.p.m.
2250 h.p.
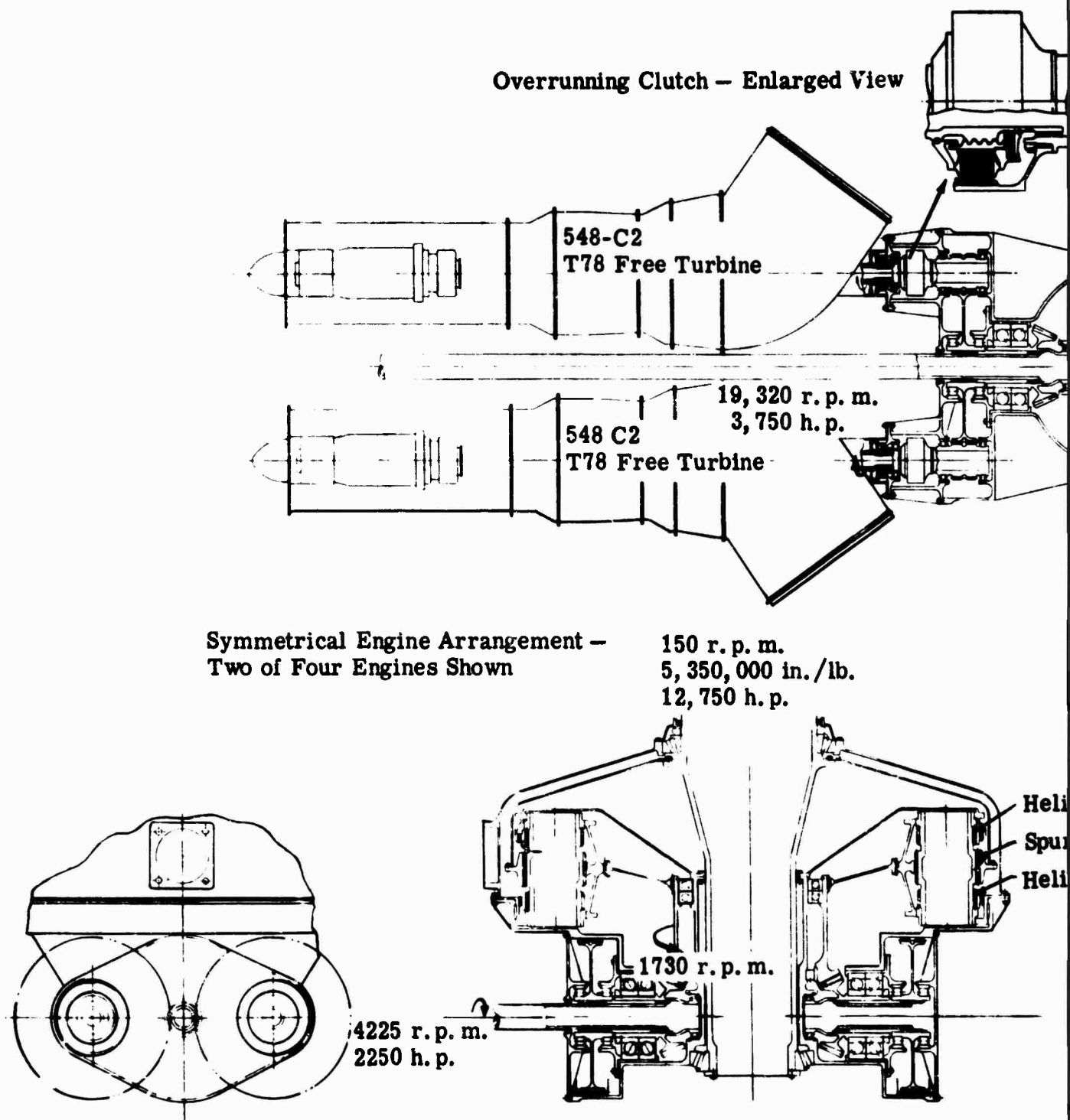
Figure 91. Combining Gearing—Parallel Arrangement Using Four Model 548-C2 Engin

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Overrunning Clutch – Enlarged View

548-C2
T78 Free Turbine

19, 320 r. p. m.
3, 750 h. p.

548 C2
T78 Free Turbine

Symmetrical Engine Arrangement –
Two of Four Engines Shown

150 r. p. m.
5, 350, 000 in. /lb.
12, 750 h. p.

Helical Gear

Spur Gear

Helical Gear

1730 r. p. m.

4225 r. p. m.
2250 h. p.

1. Combining Gearing—Parallel Arrangement Using Four Model 548-C2 Engines

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Section AA

3750 h. p.
19, 320 r. p. m.

Overrunning Clutch

4225 r. p. m.
2250 h. p.
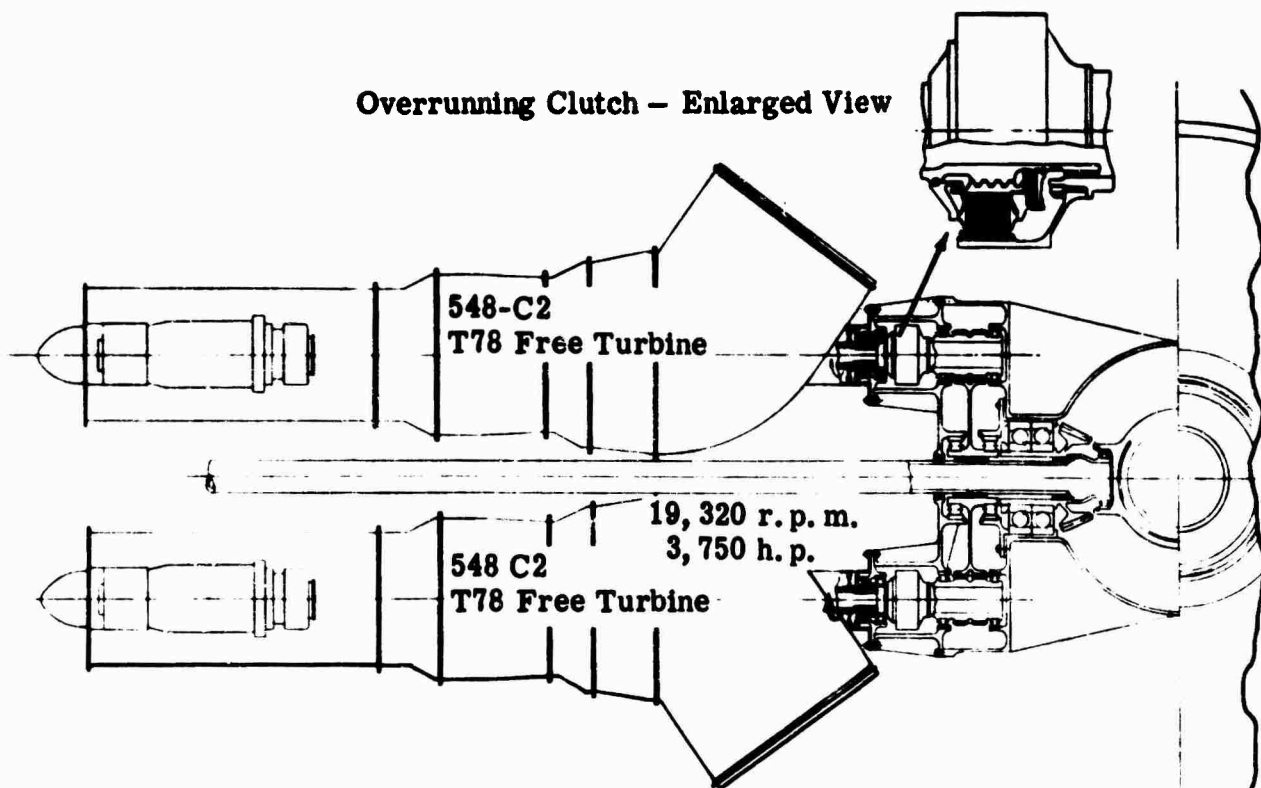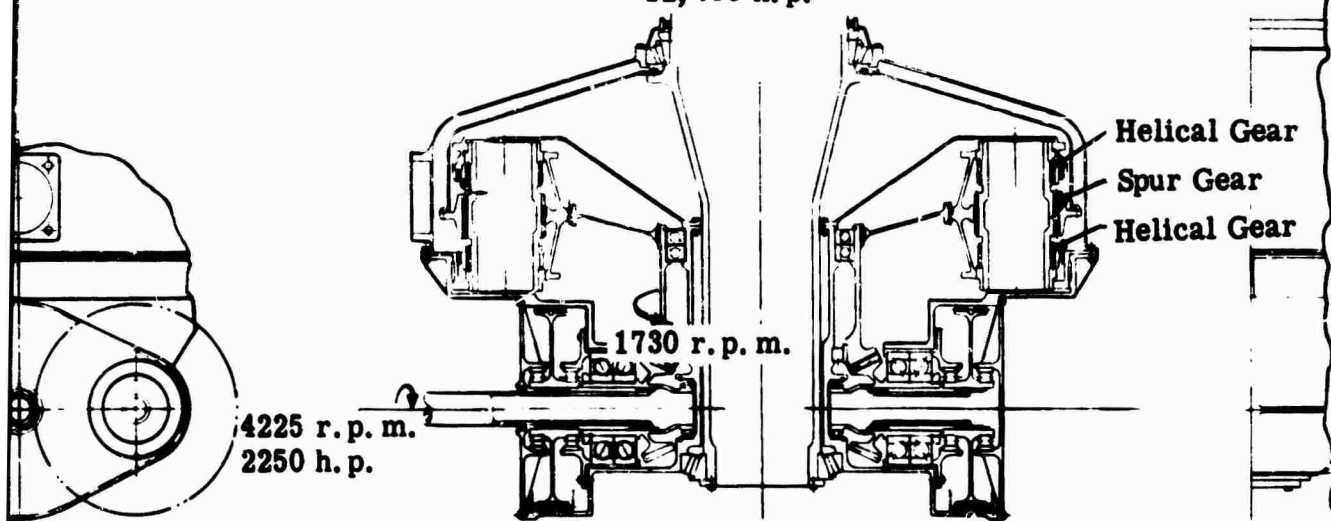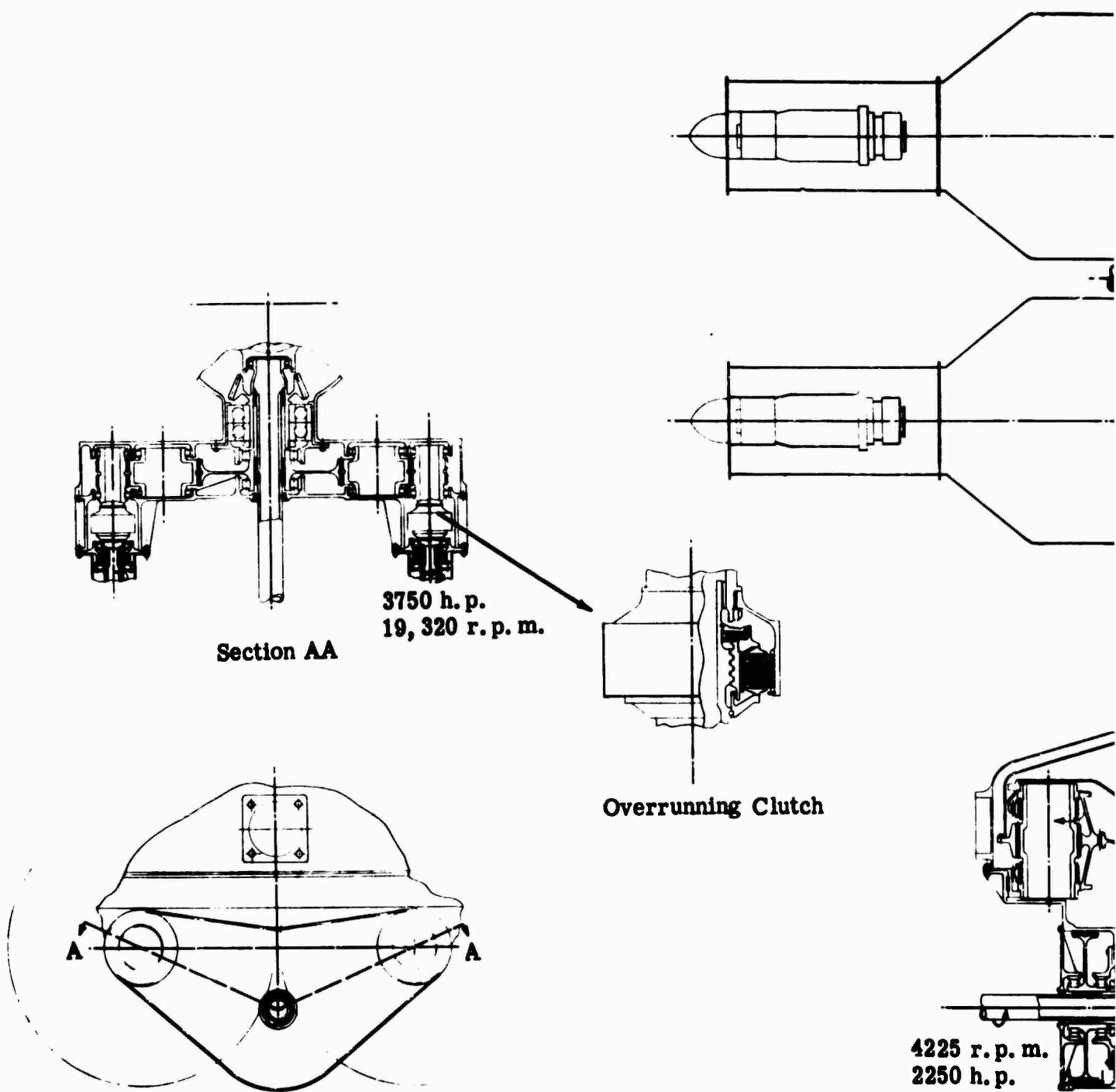
Figure 92. Combining Gearing—Parallel Arrangei

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Regenerator Free
Turbine Engine

Regenerator Free
Turbine Engine

150 r.p.m.
5,350,000 in./lb.
12,750 h.p.

Helical Gear
Spur Gear
Helical Gear

1730
r.p.m.

r.p.m.
.p.

lel Arrangement Using Regenerative Engines

2

load on the planets is reacted on planet rings that roll on corresponding rings attached to the sun pinion. At higher speeds, when centrifugal force exceeds the inward mesh reaction component, the planets roll on fixed outer rings. The weight of each planet is such that centrifugal force is nearly balanced by the mesh reaction, and, at full speed and torque, the rolling contact stress is low.

The parallel approach also could be applicable to twin rotor systems. Two engines could be arranged to drive each reduction gear assembly. The same clutches, pinions, and first-stage gears could be used while the tail rotor drive would be utilized for a cross-shaft drive. The planetary train, although identical in principle and ratio, would be sized for 7500 horsepower.

The parallel arrangement approach appears feasible; however, there are several inherent problems. The coupling of two 4500-horsepower engines so that they drive through one spiral bevel set results in high gear-tooth and bearing loading. This loading is multiplied by the reduction ratio between the engine drive and the bull gear. Arrangement of the engines fore and aft of the rotor axis can result in an exhaust ingestion problem. In addition, the transmission will be heavier than the skewed arrangement subsequently discussed. Because of these factors, further study of this approach was deemed unwarranted.

Skewed Arrangement

The skewed arrangement of the power sections allows transmission of power through individual spiral bevel gears into a first-stage ring gear. This reduces the gear and bearing loading. In addition, installation studies showed that the skewed arrangement is adaptable to both single and tandem rotor vehicles (Figures 112 and 113). Therefore, to establish the credibility of this approach from a transmission standpoint, a preliminary design investigation was made. The resulting preliminary design is presented in Figures 93, 94, and 95.

The bevel gear arrangement shown in Figure 93 provides the primary reduction from a turbine speed of 19,320 r.p.m. to 2,250 r.p.m. The bevel ring gear is driven by three bevel pinion inputs. Provision is made for a speed-increasing power takeoff for a tail rotor drive or phasing shaft. A fifth bevel pinion connects the entire reduction gear train to a rotor lock and brake, as shown.

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Sizing of the bevel gears is based on stress levels slightly less than the state-of-the-art maximum of 30,000-p.s.i. tooth bending stress and 250,000-p.s.i. surface compressive stress established by Gleason Works. Modified tooth geometry results in strength balance between the bevel pinions and the bevel ring gear. Both gear elements are straddle-mounted to maintain rigid alignment and minimum shifting of tooth contact patterns throughout the operating torque range. The arrangement was reviewed by Gleason Works, and no serious problems are contemplated on the basis of design stress levels and speeds.

A serious design problem exists, however, in the questionable scoring resistance of the high-speed mesh. Evaluation of calculated scoring parameters indicates that scoring will occur with MIL-L-7808 oil as the lubricart at any inlet oil temperature. To prevent scoring, a lubricant with higher scoring resistance is required. Evaluation of MIL-L-23699 lubricant or a high-viscosity EP petroleum oil is recommended. Initial evaluation of lubricants, platings, and coatings can be made on the Ryder rig at similar operating conditions. The overall scoring investigation should lead to full-scale testing of actual bevel gear components.

The HLH bevel gear preliminary design parameters are:

Bevel Gear Ratio	20:170
Pitch Diameter, Gear	34.0 inches
Pitch Diameter, Pinion	4.0 inches
Diametral Pitch	5.00
Pitch Line Velocity	20,200 feet per minute
Spiral Angle	35 degrees
Pressure Angle	20 degrees
Scoring Index	41,800

The final gear stage in the transmission employs a split torque gear train which was selected for this application because of the relatively high gear ratio which can be achieved combined with low planet carrier speeds and centrifugal bearing loads and the dual torque path to the rotor which permits the use of smaller output gears and bearings. The 30-inch span between radial bearings on the rotor shaft was selected from turboprop engine experience in which an 11-inch span was used with propeller 1 X P moments up to 225,000 pound-inches. It is believed that significantly larger spans would create unnecessary weight and introduce undesirable midspan deflections.
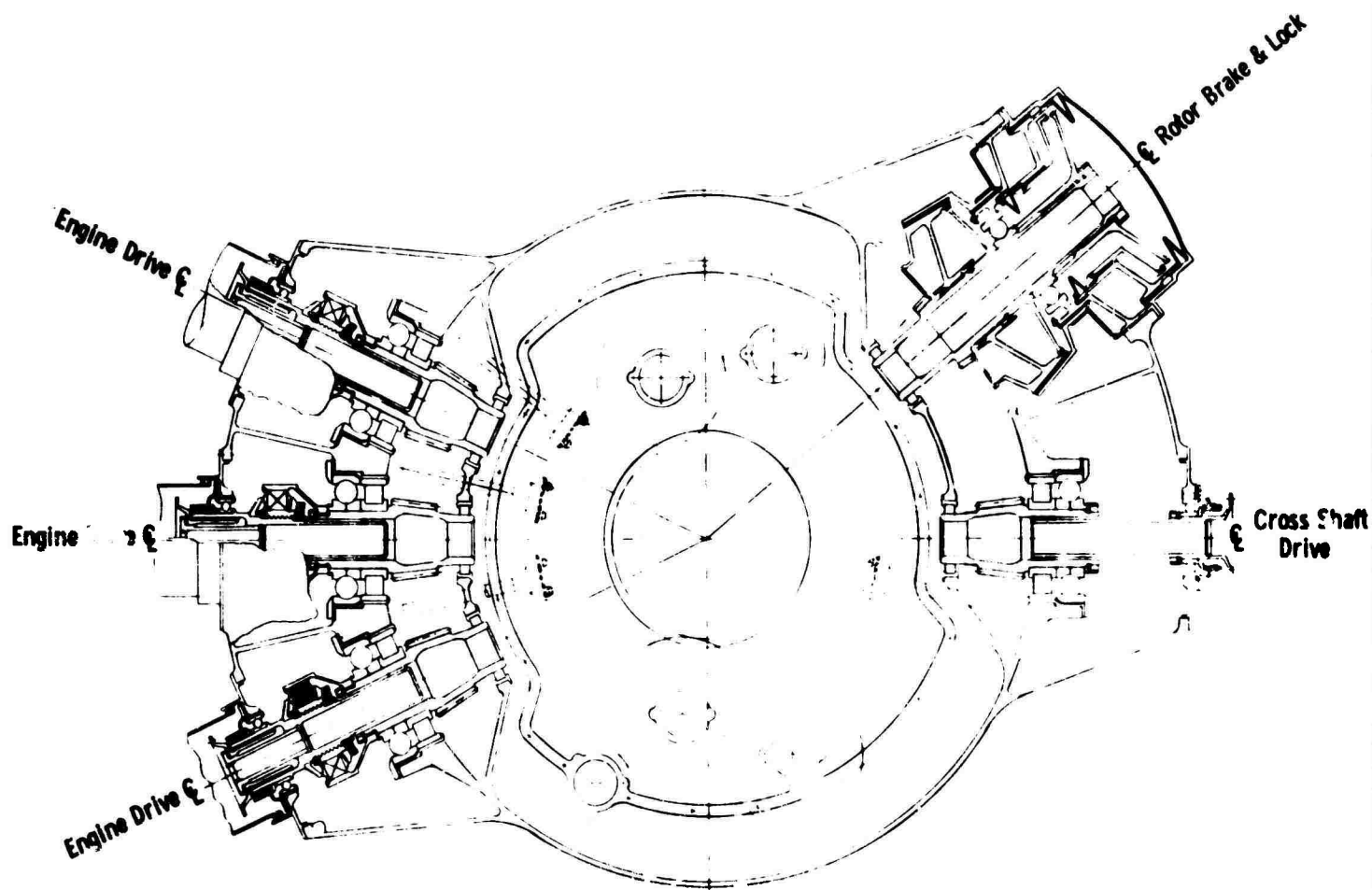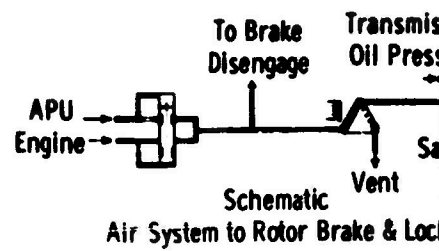
To Brake
Disengage

Transmis...
Oil Pres...

APU →
Engine →

Sa...

Vent

Schematic
Air System to Rotor Brake & Loc...

ℓ Rotor Brake & Lock

Engine Drive ℓ

Engine   ℓ

Engine Drive ℓ

Cross Shaft
Drive

Section B-B

Alternat...
Type XVI...

Figur...

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1

ssion
ssure

To Engage
Brake Lock

afety Valve

k

Relief Scavenge
Valve Pump

Cooler

Pressure By-pass Valve

Pump

Tank

Transmission

Filter Check
Valve

Pressure
Regulator

Lube System Schematic

Tachometer Pad
Type XV-B Modified Per AS 470

Alternator ₵

D

C

A

A
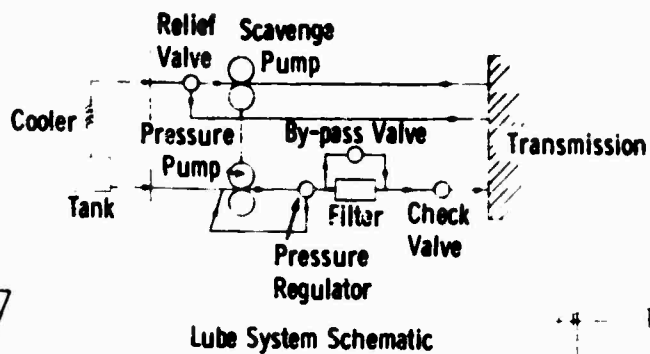
Pad
Modified Per AS 470

D

C

View Looking Up

Section C-C

93. Skewed Arrangement Transmission (SK-10059—Sheet 1)
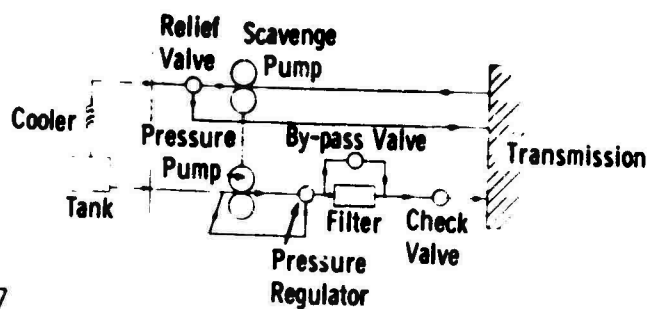
Relief Scavenge
Valve Pump

Cooler

Pressure   By-pass Valve
Pump

Tank   Filter   Check
                                  Valve
         Pressure Regulator

Transmission

Lube System Schematic

Tachometer Pad
Type XV-B Modified Per AS 470

€ Rotor

Alternator €

Section D-D

D

C

€ Rotor

A

View Looking Up   C

Section C-C

C Rotor

B

Figure 94.

Section A-A

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Skewed Arrangement Transmission (SK-10059—Sheet 2)

2

Pump

To Inlet Side
of Press Pump

Section E-E

Pump Mating Face

Section G-G

Scavenge Oil Outlet Pad —
Similar to ARP 454 A
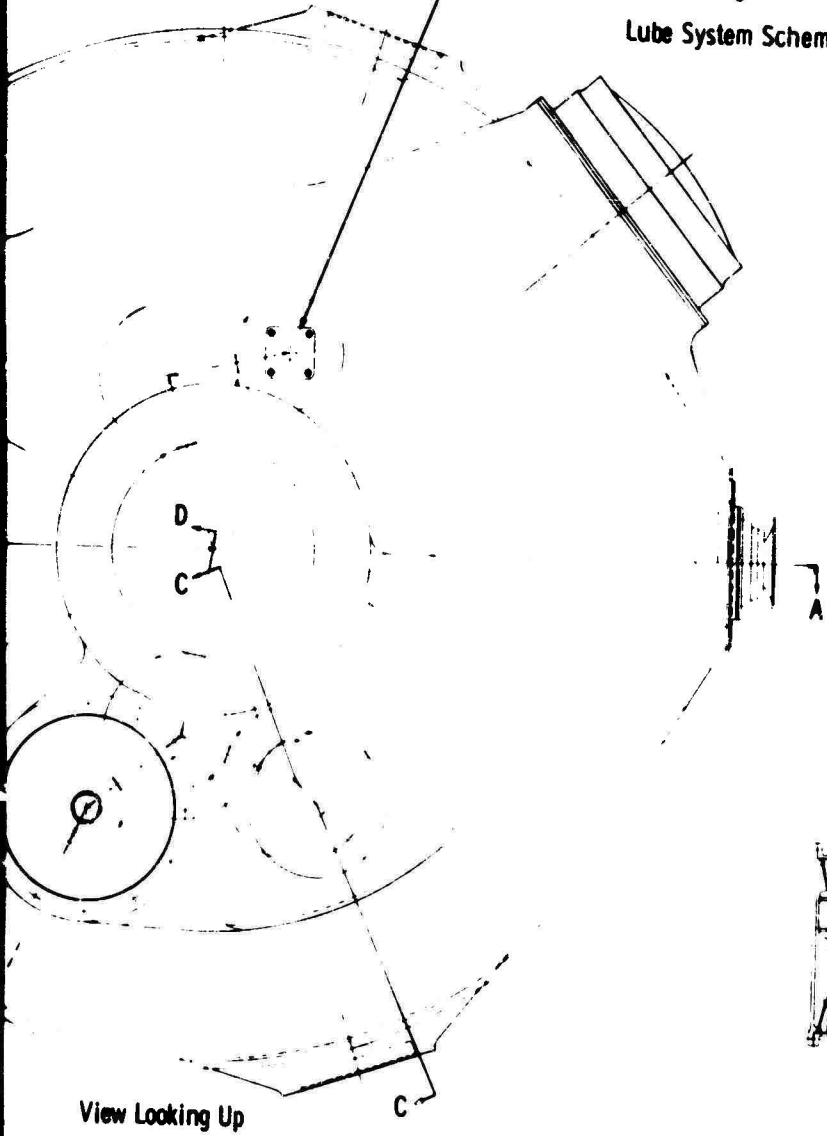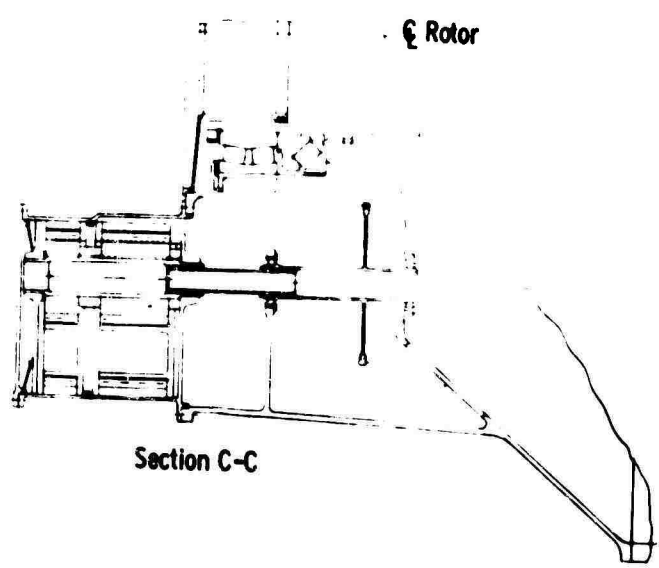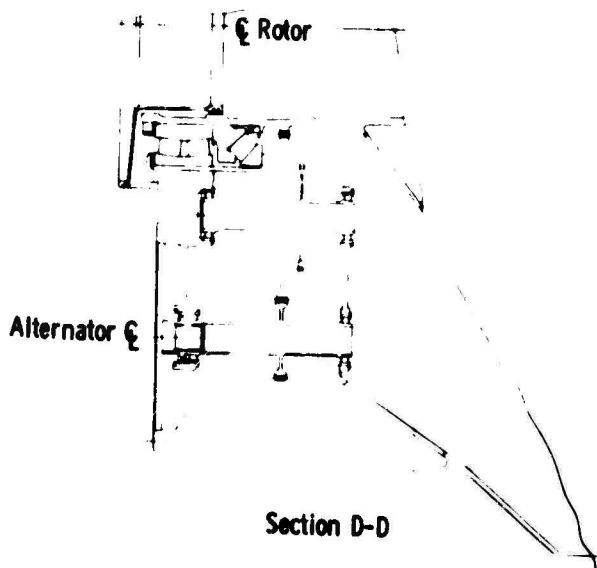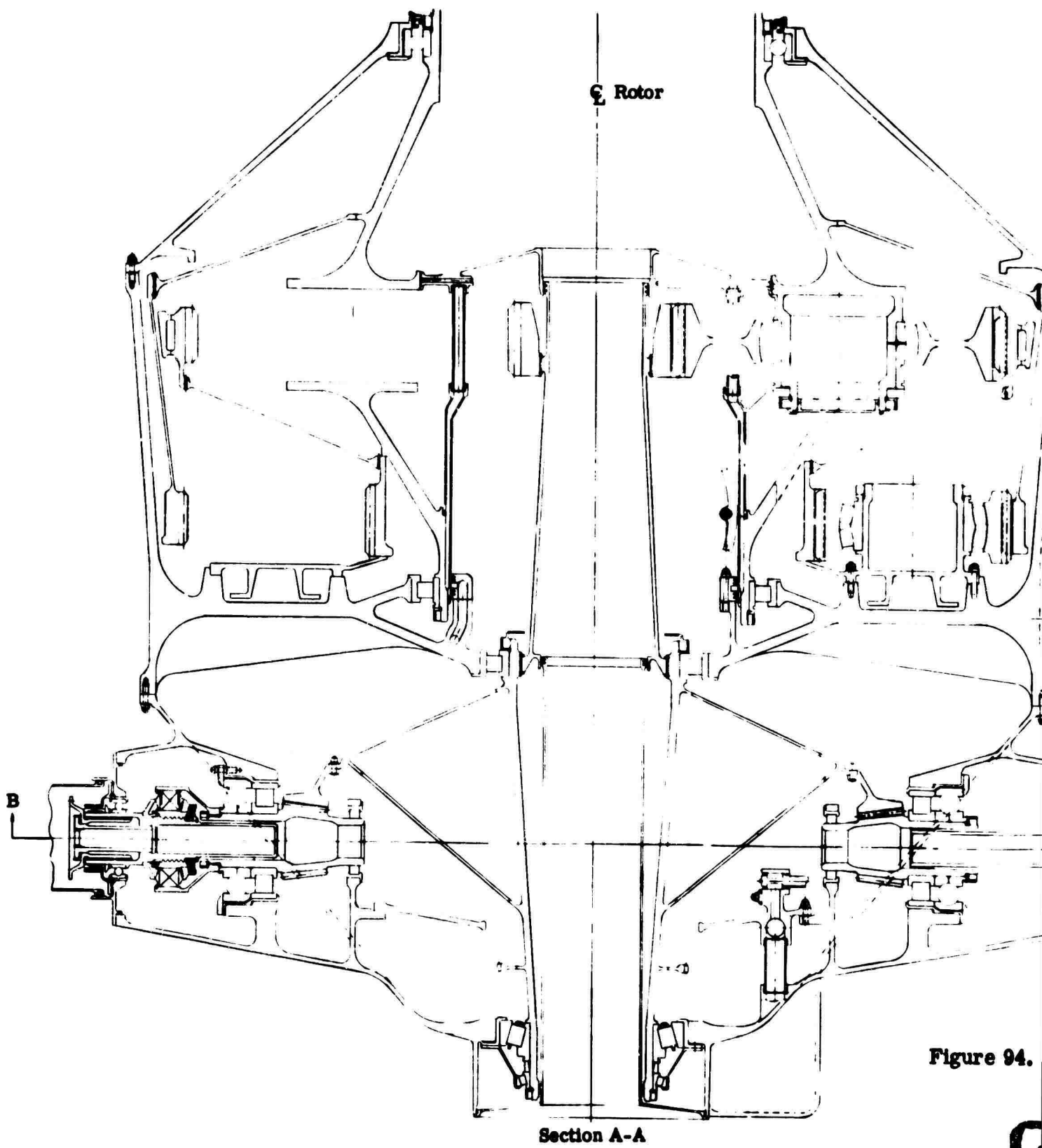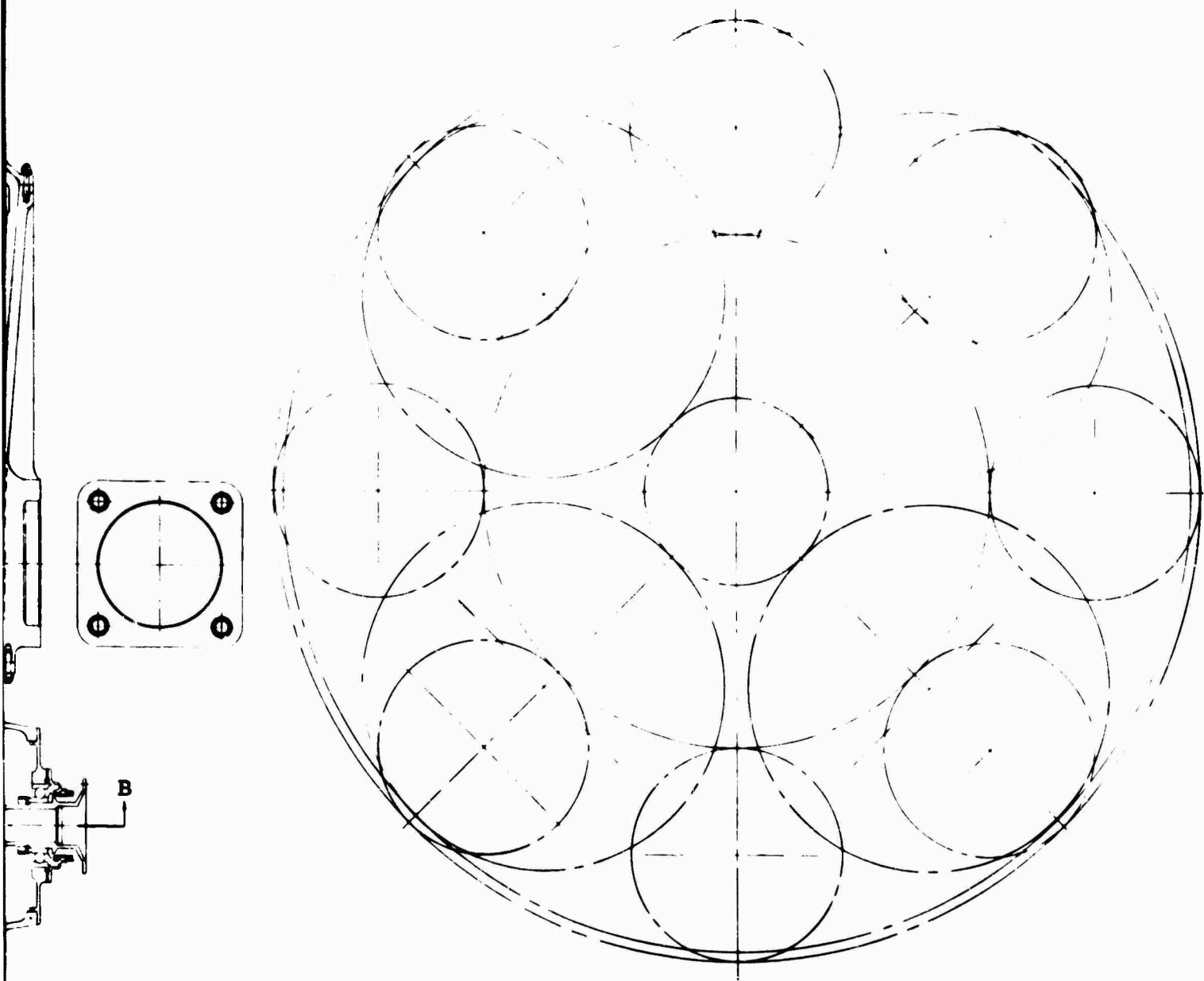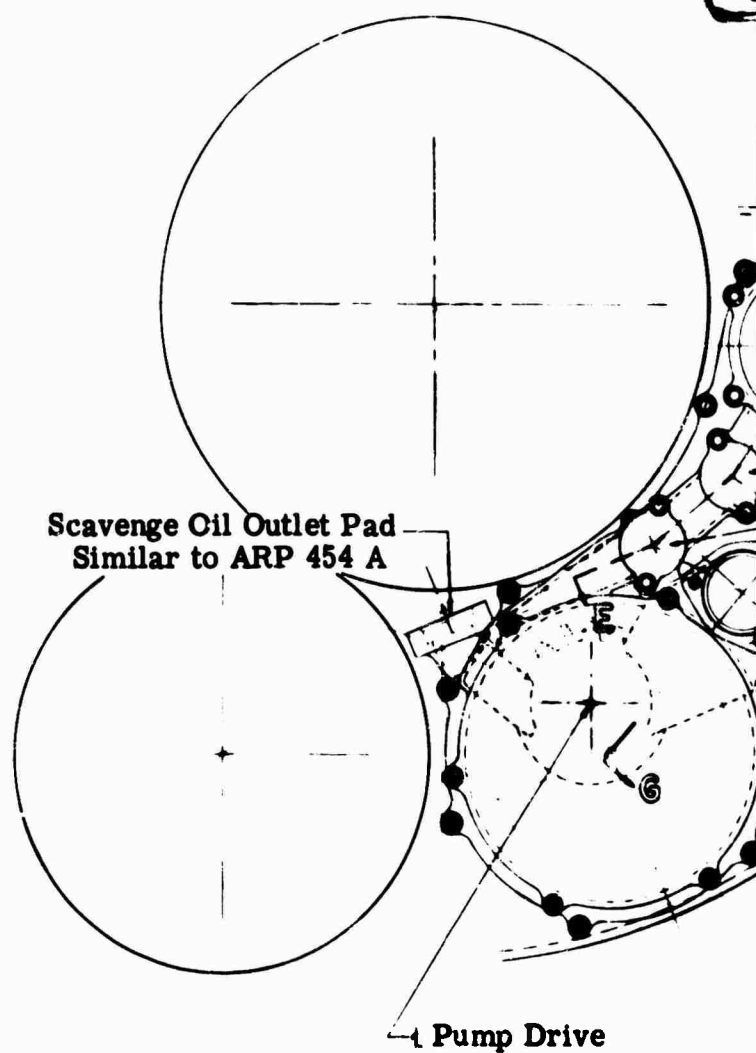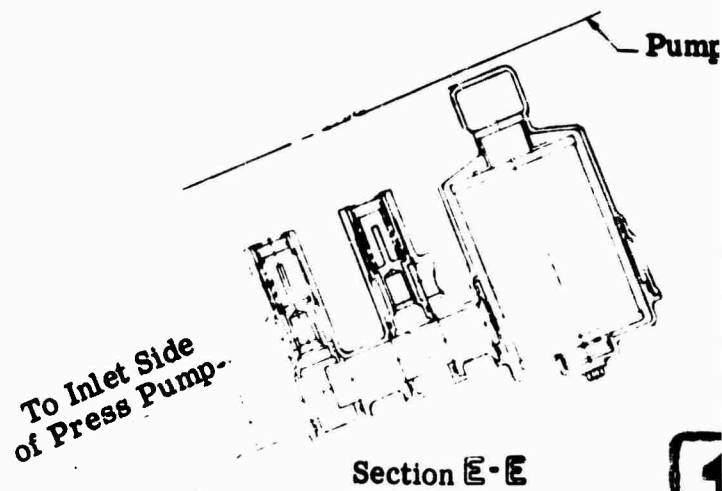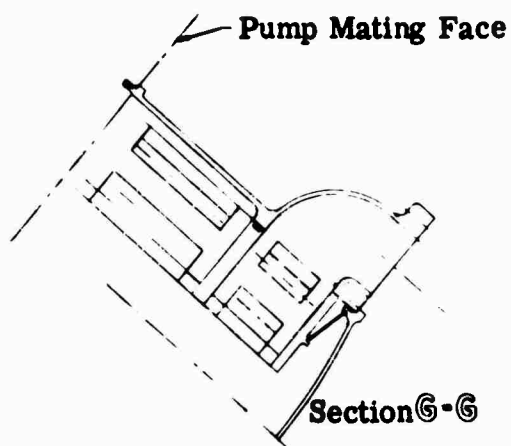
⊣ Pump Drive

Figure 95. Skewed Arrangement Transmission (SI

Pump Mating Face

Filter

Pump Mating Face

Bypass Valve

Pressure Regulator

Pressure Pump Inlet Pad
Similar to ARP 454 A

Check Valve

Check
Valve

Pressure Oil to
Transmission
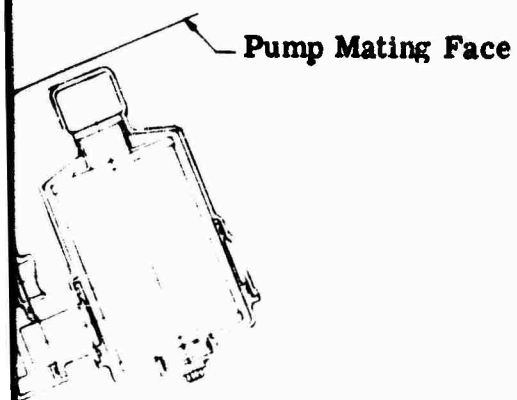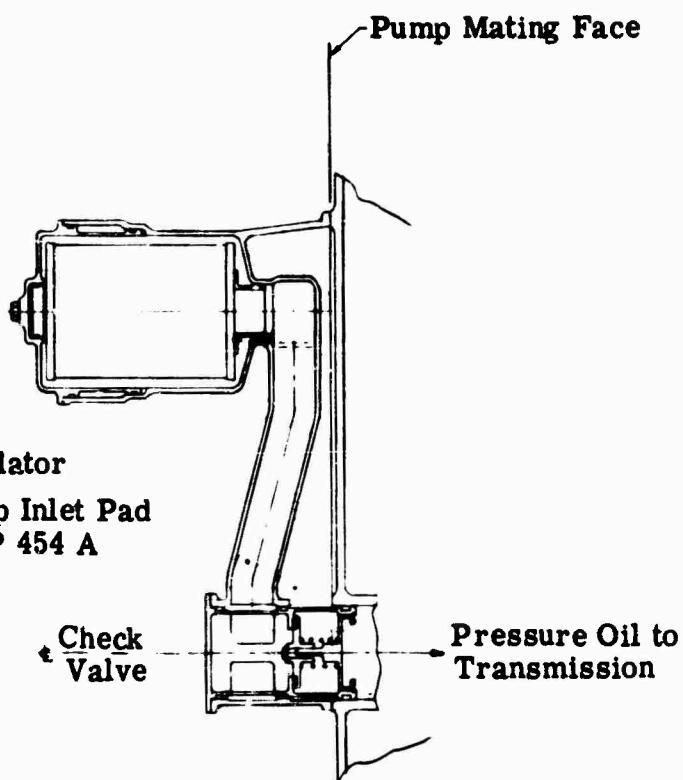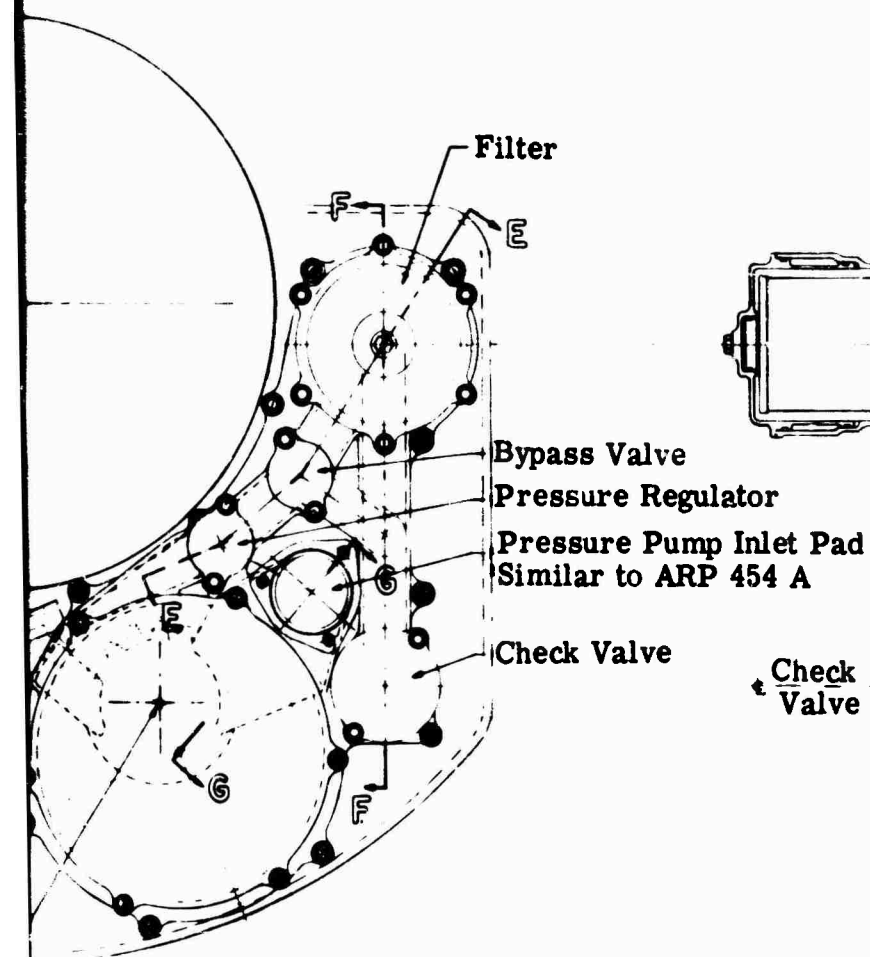
Section F-F

mp Drive

nt Transmission (SK-10059—Sheet 3)

All main power train bearings have been sized for B10 life based on Allison turboprop engine experience. These bearings, under continuous design power operating conditions, have demonstrated very satisfactory service experience in Allison commercial and military engines. The bearing parameters were also reviewed based on the mean effective power for specific missions.

Although the standard Anti-Friction Bearing Manufacturers Association (AFBMA) life calculation is used for preliminary sizing of the bearings and for comparison with experience, many other factors are considered in the final analysis. These factors are:

- Premium quality material (used in present Allison engines)
- Lubricant
- High-speed factors
- Method of lubrication

The spherical roller bearings were selected for the planet gears to isolate deflections of the carrier trunnions and rotor shaft from the gear tooth mesh and from the bearings. This arrangement is based on Allison experience with similar planet systems. Further investigation of the integral planet bearing and gear design is desired to determine bearing capacity and race deflections.

The tapered roller bearing arrangement at the bottom of the bevel drive gear was selected to permit positive location of the gear in both axial directions. This is to prevent tight gear mesh under light loads due to gear weight. Operating speeds and loads are considered within normal tapered roller bearing practices. Experience with tapered roller bearings in Allison military and commercial transmissions was used as a guide.

The cylindrical roller bearings supporting the upper end of the bevel drive gear, the bottom planet carrier (rotor shaft), and the ball bearing at the top of the rotor shaft are considered well within normal bearing practices.

A ball-thrust and two-cylindrical-bearing arrangement was selected for the input drive shafts. The high speed and the heavy thrust and radial loads dictate this arrangement. Other arrangements studied were as follows.

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● A tapered roller bearing arrangement would be desirable to provide positive axial location for optimum mounting of the bevel pinion. The 1,400,000 DN value, however, is well above normal limits for this type of bearing. The flange friction inhei ent in this type of bearing is also undesirable.

● A duplex ball bearing was considered to further reduce centrifugal force loads; however, a computer analysis indicated that the outer race Hertz stress was less when a single thrust bearing was used. Inner and outer race differential contact angles and shoulder heights are within acceptable practice for thrust bearings. A single ball thrust bearing was selected due to the cost, reliability, and assembly advantages.

The larger roller bearing on the bevel pinion drive shaft was analyzed by IBM computers to evaluate centrifugal force loads. This study indicated that the bearing life was still controlled by the inner ring life. Therefore, normal bearing life should be achieved with normal distribution among the bearing components.

The smaller input shaft roller bearing and the small ball bearing should be closely evaluated due to the high speed. Skidding of high-speed roller bearings has in the past contributed to excessive bearing and related gear failures.

The lubrication system is arranged to provide accessibility to all pumps and valves. A regulated system pressure of 50 p.s.i. is recommended for the transmission to permit the use of large jet holes and to minimize the danger of plugging. One exception to this rule is the high-speed pinion mesh where a pressure of 180 p.s.i. is required to obtain penetration into the gear teeth as they come out of mesh. The lubrication system schematic is shown in Figure 93.

The transmission includes a unique rotor brake and lock. This device is discussed under the subsection titled Safety Devices. Note that this device, as well as pressure regulators, pumps, and filters, can be readily removed for service without disturbing the transmission or the installation. This feature will reduce maintenance time,thereby increasing flight availability time.

CROSS-SHAFTING

Present planning by industry for the HLH indicates the need for cross shafting over a span of approximately 60 feet to connect tandem rotors

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or drive an antitorque rotor. Past and present practice has been to design such shafts to operate below their first critical speed by using support bearings spaced at intervals determined by the operating speed of the shafting. Larger vehicles which require longer shafting will use more bearing supports with a corresponding reduction in system reliability. As shown in Figure 96, the weight of a simple shaft designed to transmit a given power will decrease with increasing speed. However, when bearing supports, are added to the shafting, the combined subcritical system weight is a minimum near 6000 r. p. m. and becomes heavier at the higher speeds, as shown in Figure 96. To take advantage of lightweight, high-speed shafting, it is desirable to operate at supercritical speeds by removing some of the intermediate bearing supports. A very significant amount of theoretical and subscale test work has been accomplished at the Battelle Memorial Institute under the sponsorship of USAAML, showing that supercritical designs may be feasible for cross shafting in the HLH. Accordingly, Figure 97 was prepared using information from the Battelle studies to show mechanical features which are considered desirable. First, although the total span may approach 60 feet, it is believed two 30-foot lengths should be considered, to simplify manufacturing, transporting, and replacement in the field. Aluminum shafts can be extruded in the required length economically and with close control of variations in wall thickness. All bearings and shafts should be designed for individual service and replacement without disturbing transmission installations, and should be oil lubricated to simplify servicing and to improve reliability at the higher supercritical speeds.

Close coordination between airframe and shafting designers will be required to ensure that deflections, spring rates, and damping factors of the two systems are compatible with each other and with exciting forces created by rotor blade passage and engine disturbances.

The shafting system shown in Figure 97 represents a generalized approach which could be used in a typical installation. Specific design features would be modified to adapt the system to each particular airframe. For instance, the first shaft mode falls at 12 cycles per second. This frequency would be unsuited for use with a six-bladed rotor operating at 120 r. p. m., due to the coincidence of rotor blade passage frequency with the fundamental shaft resonance. Airframe bending mode; can lie in the range of 5 to 30 cycles per second; the shaft design should avoid placing the lower frequency modes near major fuselage bending frequencies.

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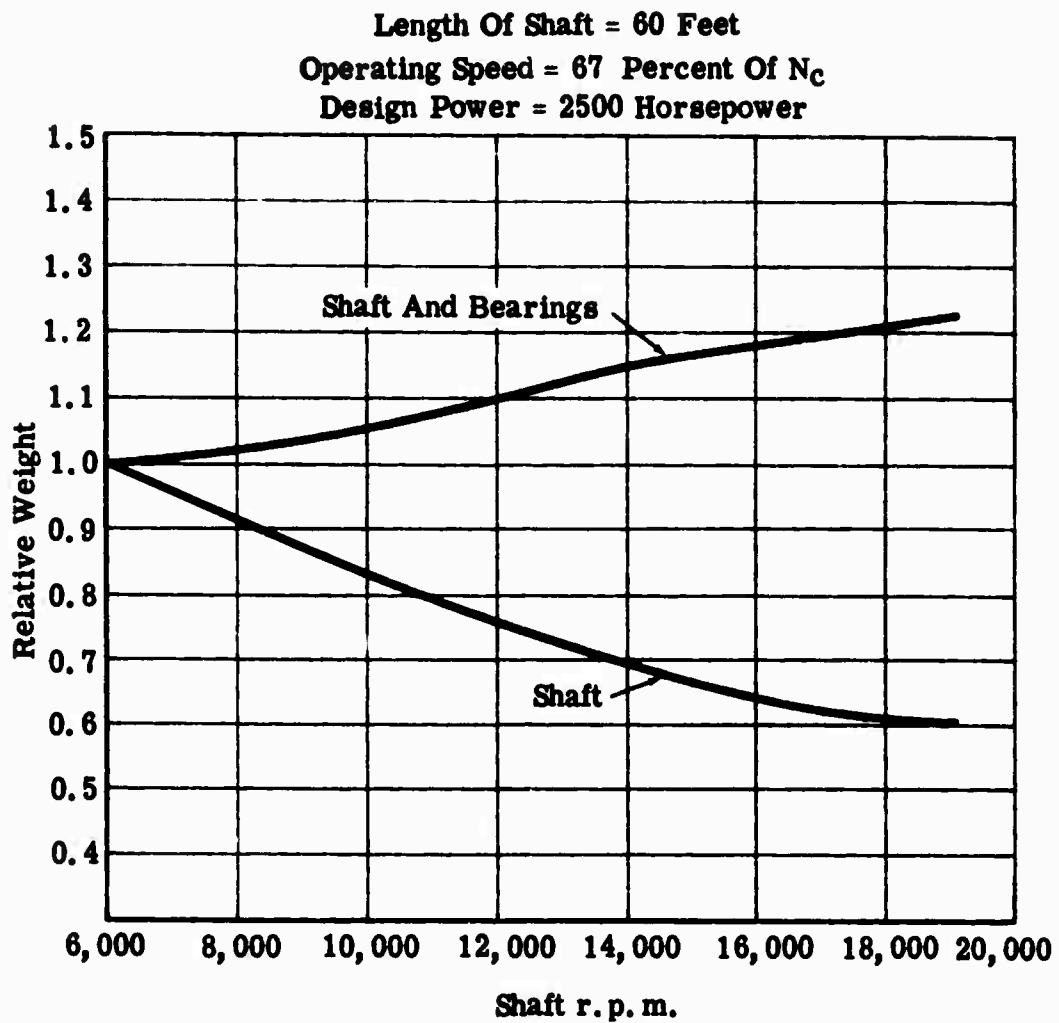
Figure 96. Subcritical Shaft Weight

The shaft is designed so that it operates below the nineteenth mode
critical speed at design speeds up to 20,000 r.p.m. The two damper
bearings on each of the shaft sections should suffice to dampen all shaft
modes encountered '.. this speed range. Placement of these bearings is
based on the work performed at Battelle Memorial Institute. One damper
is at the first antinode for the fourth mode and will be effective primarily
in damping the lower, simpler modes. The other damper is at the first
antinode for the twelfth mode and will dampen the higher frequency modes.

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C Support

Fig. No. 1

Fig. No. 1

Fig. No. 2

60 ft

C Support Bearing

C of Main Rotor Shaft

2.0 Max. Allowable Axial Travel

7.500

Fig. No. 2

Air Inlet 5 Inches of Mercury Gage Type 4 Places

19,300 r.p.m.

Rubber Washer

Belleville Spring

Oil Level

Fig. No. 1

View Looking Into Shaft Bearing Support Housing With End Plates Removed, Also Section Thru Mtg. Typ 4 Places

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Figure 97. Cross Shaft
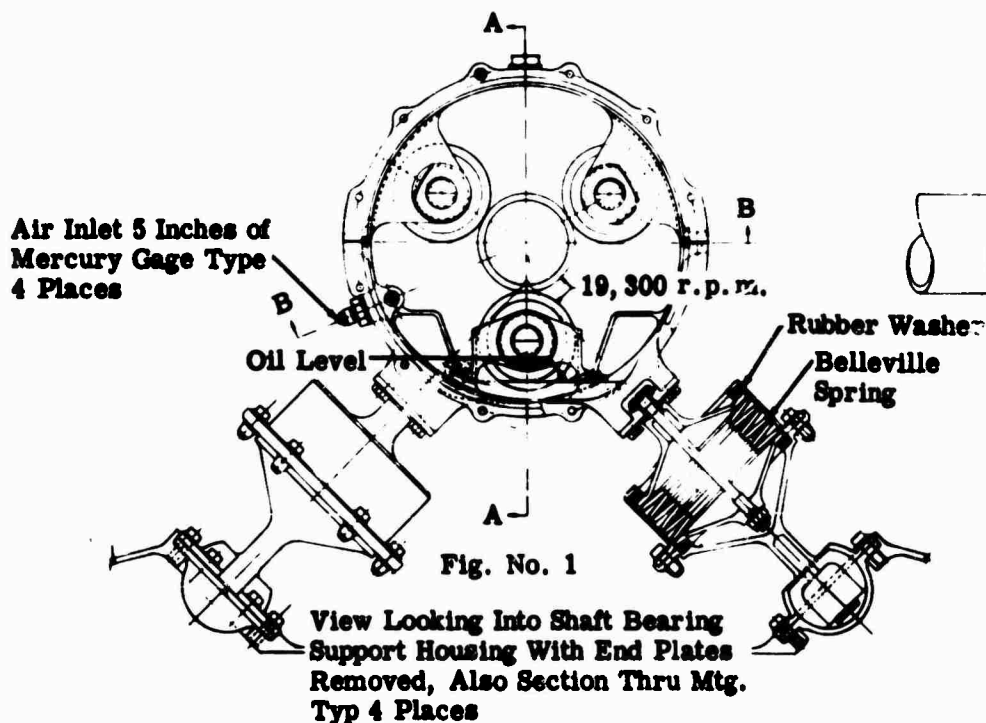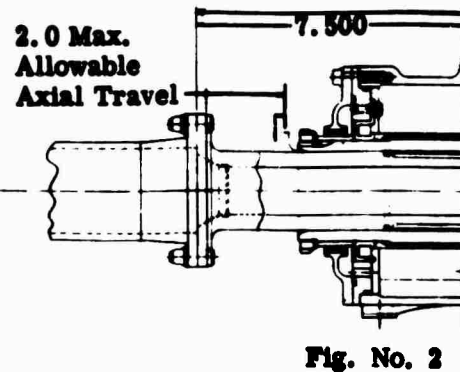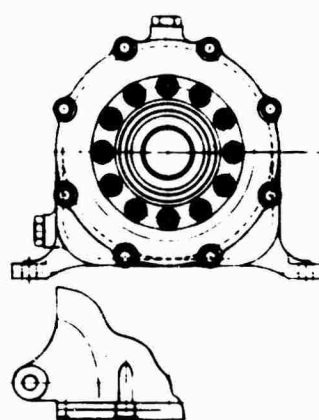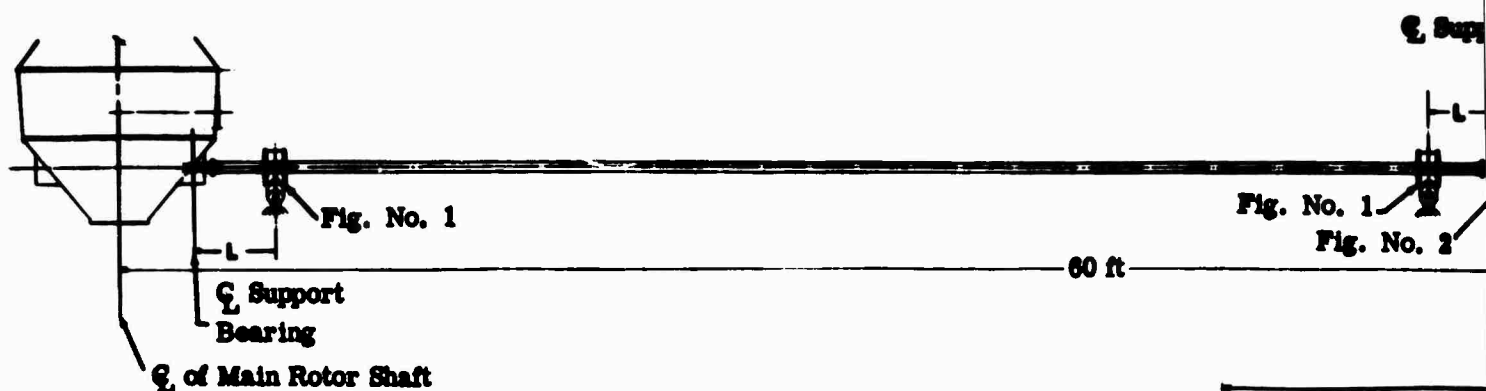
₵ Support Bearing

₵ Support Bearing

No. 2 30 ft.   Fig. No. 1   Scale One Tenth Actual Size   Fig. No. 1

—15.00—

Extruded Aluminum
Shaft Approx. 328.5 Long
2 Places Typ.

—2.15 ID

—2.50 OD

Oil Level

. No. 2 Ball Spline Coupling

Air Outlet

Molded Hard Rubber
Seal (Split)

Flame Plate Shaft
in This Area

isher

Oil Level

Design Data
Design Steady State Horsepower—2500
r.p.m.—19,300
Shear Stress—p.s.i.—6,000
Windup at 19,300 r.p.m.—0.218 Rev.
Windup at 1,000 r.p.m.—0.0113 Rev.

Air Inlet

Section B-B

Section A-A

ss Shafting (AL-15301)

The spring rates and damping coefficients should be selected as described in the Battelle report.* This selection will be influenced by the airframe dynamic characteristics and thus has not been accomplished on this preliminary design. In addition, Allison test experience has shown that there is a good possibility that shaft resonances can be controlled by attaching a nonrotating inertial mass to the shaft at the proper points. Dynamic analysis of this concept may lead to more optimum configurations.

The shaft is sized on the basis of minimum weight for the required torque loading, with a wall thickness that will not present great difficulty in fabrication in the desired lengths. Further evaluation of the shaft design would include studies of trade-offs between wall thickness, weight, manufacturing costs, handling fragility, and buckling potential. Improvements in beryllium technology may make the use of this material feasible if costs are not prohibitive.

The design includes a fixed midspan bearing. This feature tends to decouple the two sections of the shaft and thus reduces by half the number of vibrational modes by eliminating symmetric and unsymmetric modes in which the midpoint would act as a simple support. Experience has shown that conventional splines are incapable of sliding while transmitting high torque. Airframe deflections and thermal effects will require the shaft system to accommodate axial motion. As shown in Figure 97, a ball spline located at the midbearing pillow block and a second spline (not shown) at one end of the shafting will allow axial motion while transmitting torque. Flexible couplings used in many current installations are not considered necessary with supercritical shafts of this type because of their relatively low spring rate and correspondingly reduced bearing loads resulting from shaft deflections. Static deflection of the illustrated shafts would be approximately 1 inch at midspan. The corresponding value for a 60-foot shaft would be 16 inches, assuming simple supports at the ends. When vertical gravity loads are introduced it will be necessary to consider airframe clearances which are appreciably larger than those of current practice.

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*R. G. Dubensky, C. C. Miller, Jr., and J. E. Voorhees, Design Criteria For High-Speed Power-Transmission Shafts, Technical Documentary Report No. ASD-TDR-62-728 (Part 1). Flight Accessories Laboratory. Directorate of Aeromechanics, Aeronautical Systems Division, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio, January 1963.

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Damper bearing housings are split horizontally, for easy bearing inspection, and are provided with small oil sumps to provide splash lubrication without adding excessive unsprung mass to the shaft. The aluminum shaft is flame-plated with tungsten carbide in the bearing track to provide a wear-resistant surface. For long service and to avoid shaft damage, nonrubbing rubber labyrinth seals are used to retain oil in the housings. A very small amount of compressor bleed air at a pressure of 6 to 10 inches of water would be required to create a favorable pressure differential across the seals. Airflow into the housings would be vented overboard through a simple oil centrifuge. To ensure intimate contact between each set of damper bearings and the shaft, one bearing is spring-mounted so that a predetermined preload can be applied when the housing shell is bolted together. This, or a similar feature, is considered essential in transmitting damping loads into the shaft without excessive vibration and shaft wear.

A preliminary comparison between the weight of a conventional subcritical shaft turning 6,000 r.p.m. and the 19,300-r.p.m. shaft being described is presented in Table 12.

TABLE 12
COMPARISON OF CROSS-SHAFT WEIGHTS

Material	Steel	Aluminum
Speed, r.p.m.	6,000	19,300
Length, Feet	60	60
Horsepower	2,500	2,500
Torque, Pound-Inches	26,200	8,150
Outside Diameter, Inches	2.5	2.5
Inside Diameter, Inches	2.3	2.15
Shear Stress, p.s.i.	30,000	5,930
Shaft Weight, Pounds	192	94
No. of Bearing Supports	15	5
*Bearing Weight, Pounds	120	50
Total Weight, Pounds	312	144

*Weight of bearing support structure not included.

From the preliminary calculations, the empty weight of the HLH can be reduced by at least 168 pounds if supercritical shafts are used in single rotor designs. The weight reduction in dual rotor machines will be somewhat larger due to the greater power which must be transmitted and the

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increased size of the associated components. Additional weight savings should also be realized when airframe-furnished support points are included in the analysis.

The effect of torsional windup was considered for a series of steel shafts designed to transmit 2,500 horsepower at speeds from 6,000 to 19,300 r.p.m. Wall thicknesses were arbitrarily set at 0.1 inch as the lightest design in steel which could be manufactured economically and provide long life in the field. Shaft diameters were selected to provide a shear stress of 30,000 p.s.i. As shown in Figure 98, shaft windup increased in the smaller, faster shafts. However, when these windups were converted through increasing gear ratios to a common output speed, the windup at the rotor decreased with higher cross-shaft speeds. Therefore, it would appear that better timing between dual rotors can be achieved with high-speed cross-shafts.

In summation, it is believed that significant weight can be saved and system reliability improved by using supercritical cross-shafting in the HLH. An extensive analytical and full-scale test program will be required when a vehicle configuration is established to demonstrate compatibility between the shafting and airframe systems.

CLUTCH ANALYSIS AND DESIGN

Hydraulically actuated disk clutches were studied for fixed shaft engines. From this study, it is concluded that a friction-type clutch is feasible and that it can be developed into a reliable piece of equipment.

The following characteristics of the rotor were selected as representative for study purposes:

- Minimum air drag or power required to turn the rotor at rated speed (150 r.p.m.) is 1500 horsepower.
- Polar moment of inertia at the rotor shaft is 130,000 slug-feet squared.

Preliminary investigation revealed the desirability of having a clutch assembly on each engine drive shaft rather than one large clutch in the reduction gear. This arrangement would minimize the individual size of the clutch and provide greater reliability and versatility of operation and installation.

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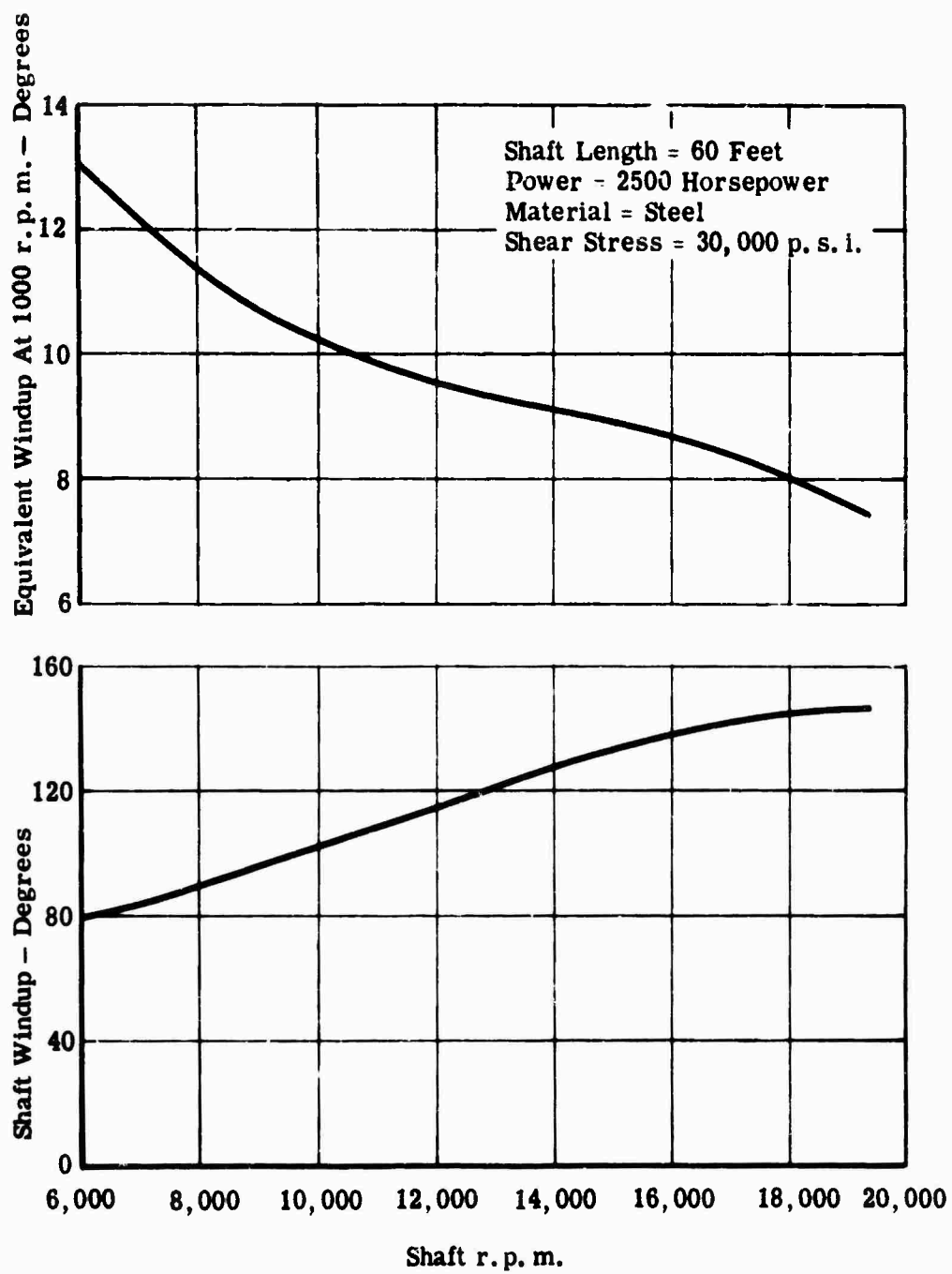
Figure 98. Shaft Windup as a Function of Shaft r.p.m.

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The ability to make a complete rotor acceleration and clutch engagement in less than 30 seconds using three engines was selected as a design target. The major parameter determining the size and cooling oil flow rate of a clutch of this type is the rate of heat dissipation. Accordingly, the clutch size and cooling oil flow rate can be optimized by scheduling the slip torque versus output speed or slip. This can be accomplished conveniently by (1) having the clutch control pistons on the output side of the clutch and (2) designing to make use of the centrifugal head pressure developed in the clutch engagement chamber as output speed increases. A clutch torque characteristic compatible with the use of centrifugal head pressure was optimized as shown in Figure 99. The designed torque characteristic meets the clutch time design target and has a maximum slip torque below the hot-day power available characteristics of the Model 545-C2 engine. Ample margin is provided to allow for clutch variation and engine deterioration.

Rotor acceleration characteristics are shown in Figure 100. Three curves were determined, based on engaging two, three, and four clutches simultaneously. With the output side of the clutches geared together to the rotor and the torque characteristics being a function of clutch output speed, all engines would automatically share the load during an engagement.

Two curves on heat rejection rate are shown in Figure 99. One is based on the engine clutch input speed being held constant at 100-percent rated speed and the other based on reducing the engine speed to 90-percent rated speed during the slip engagement period. By integrating these curves with the acceleration time curves of Figure 100, the total heat rejection can be determined.

As the air drag torque is a relatively small portion of the total clutch torque, the total heat rejection does not change substantially with the acceleration time associated with number of engines or clutches used. It would be somewhat smaller as the time is reduced. The total heat rejection associated with the three-engine engagement was calculated to be 22,835 B.t.u. with 100-percent speed input and 17,930 B.t.u. with 90-percent speed input. It can be seen that even though the maximum heat rejection rate is reduced only 10 percent, total heat rejection is reduced by more than 21 percent. A cooling oil flow rate of approximately 220 pounds per minute per clutch or a total of 660 pounds per minute for the three clutches is required for 90-percent input speed engagements. When it is considered that the rotor reduction gear would require over 500 pounds per minute oil flow for normal lubrication,

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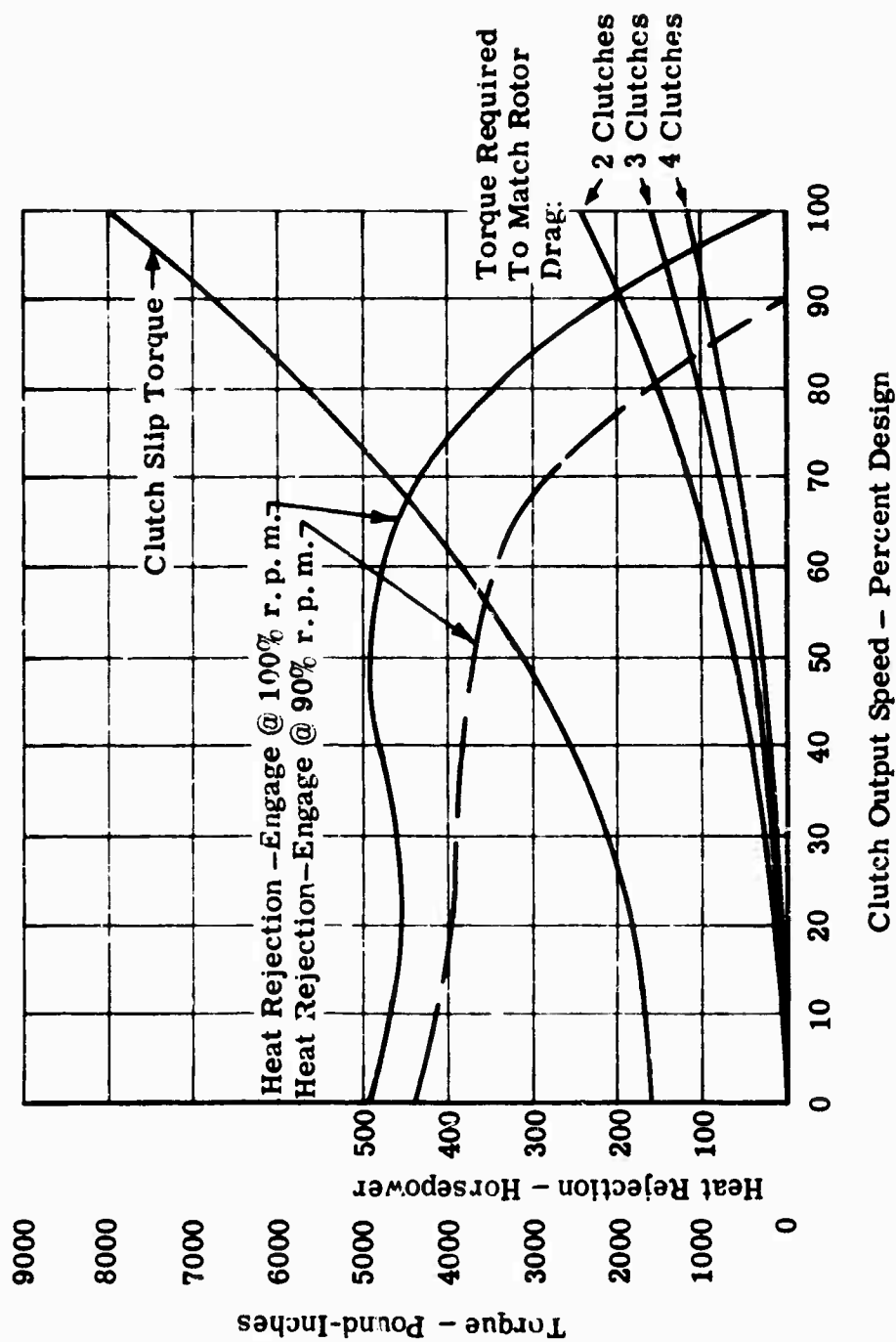
Figure 99. Torque and Heat-Rejection Characteristics
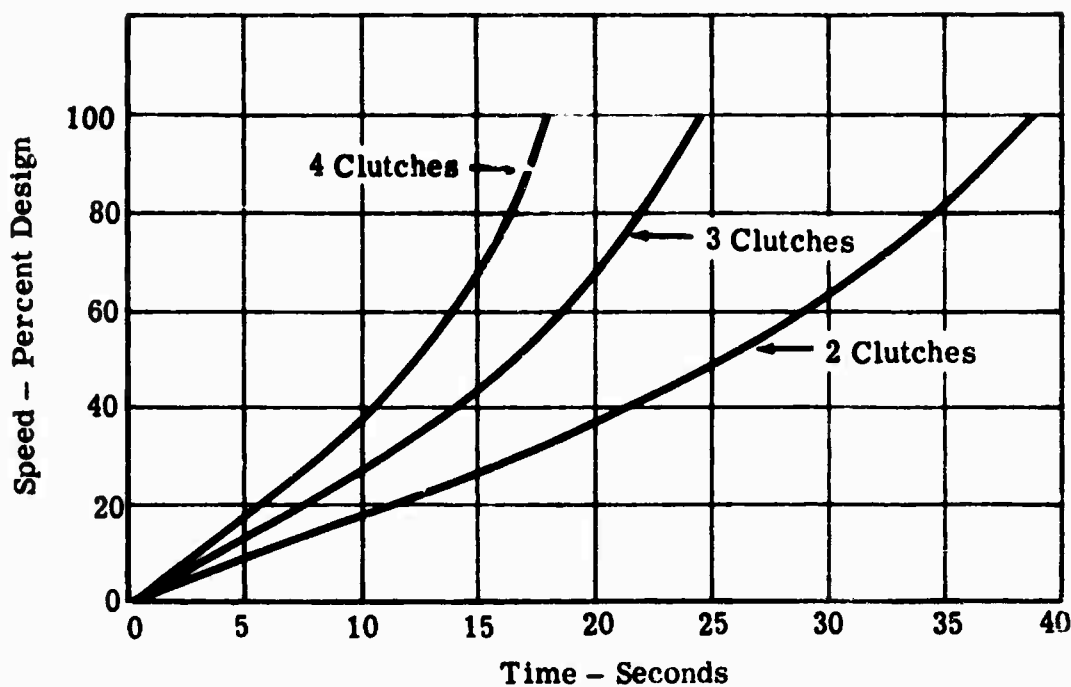
Figure 100. Acceleration Characteristics

extra pump capacity for the clutches can be minimized by diverting most of the gearbox lubrication flow for clutch cooling during the brief slip period. This can be accomplished automatically by a valving system, as shown schematically in Figure 101.

With the high oil flow rate of the rotor reduction gear and its cooling requirements, it is considered that no substantial increase in cooler capacity would be required. The lube oil reservoir would serve as a heat sink with a temporary increase in lube oil temperature. The 17,930 B.t.u. expended during the rotor engagement is equivalent to raising the temperature of 344 pounds of oil approximately $100°F$.

With the aforementioned requirements and based on experience with high-speed clutches, a clutch assembly was designed that is compatible with the high rotational speed (19,300 r.p.m.) of the Model 545-C2 engine. (See Figure 102.) The calculated weight of this assembly is 53 pounds and it incorporates features which would enhance reliable service operation.

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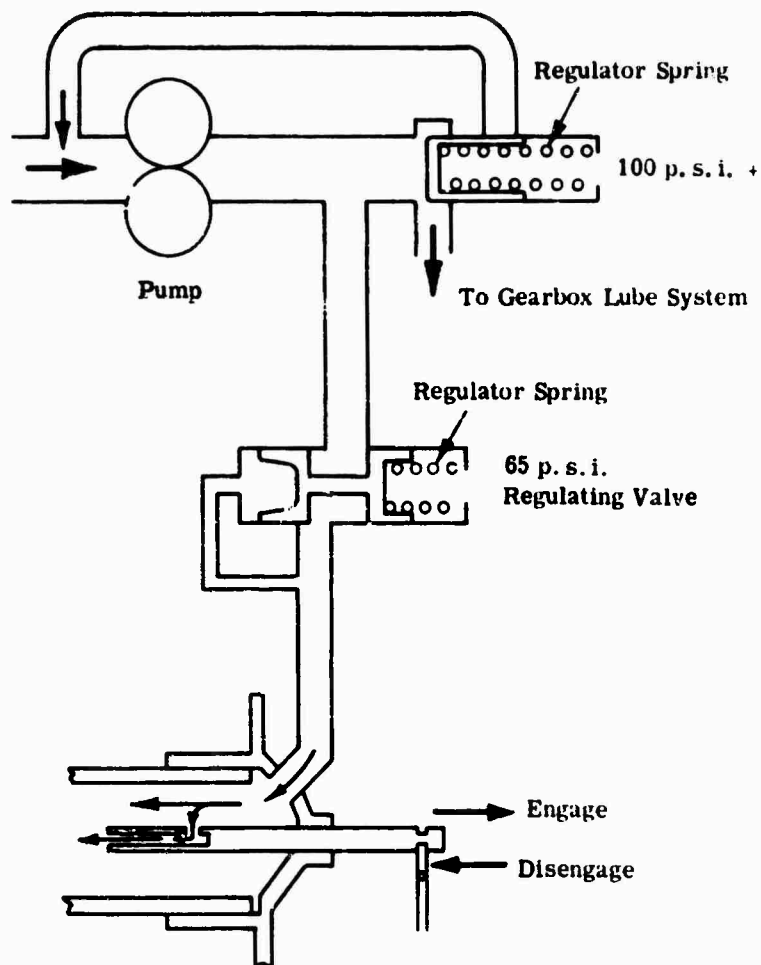
Figure 101. Valve System for Clutch Control

To avoid prolonged disengaged running of the clutch plate pack, a positive engagement clutch with a cone synchronizing blocker clutch is in series with the main clutch. This device serves two purposes.

1. It accelerates the main clutch outer member and controls its engagement.
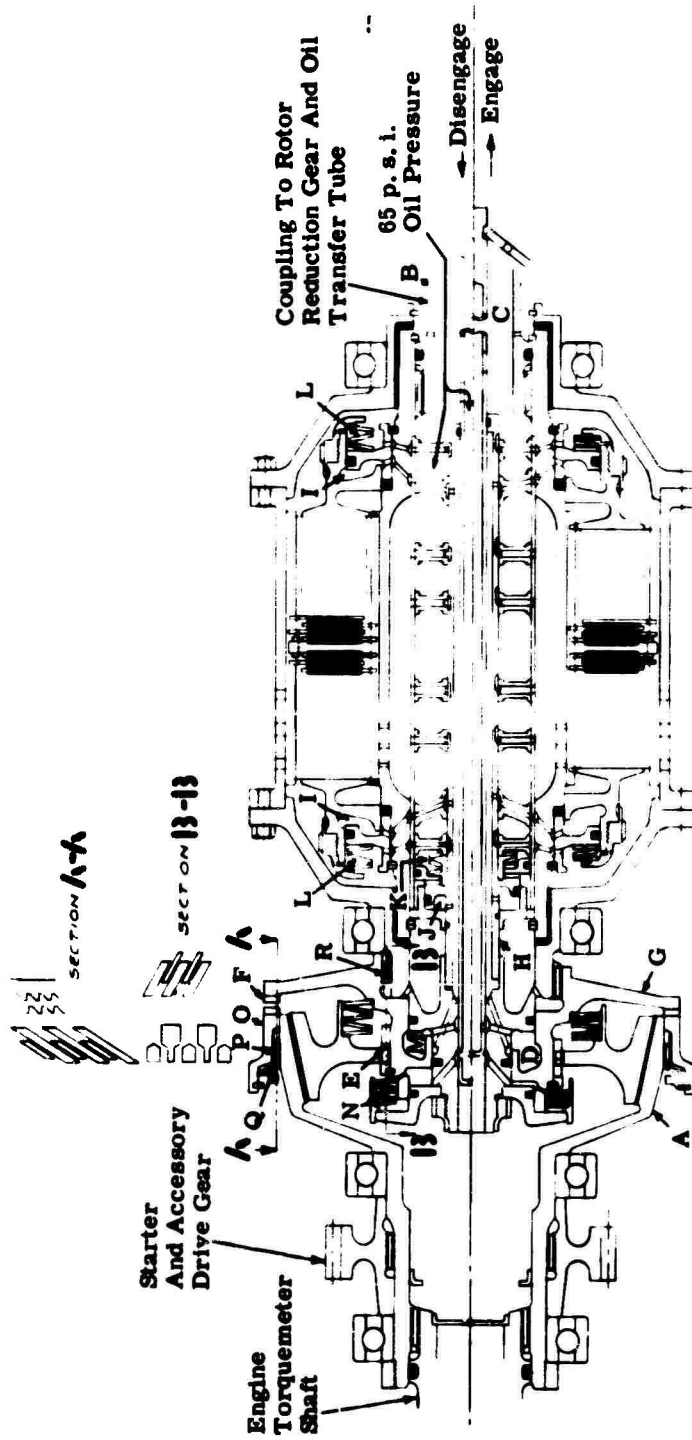
Figure 102. Clutch Preliminary Design

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2. It is sensitive to reverse torque and disengages automatically
    in case of engine malfunction or programmed underspeed,
    the.eby replacing the need for a separate overrunning clutch.

The torque characteristics of this device are shown in Figure 103.
The main clutch outer member is accelerated to synchronous speed in
less than 2 seconds by this torque. This is additive to the time, as
shown in Figure 100. The drag torque range in the reverse direction
for automatic disconnect is also indicated. This low drag torque is
eliminated when the device is energized to the disengaged position.
Under this condition, the cone clutch is completely separated and there
are no rubbing surfaces.
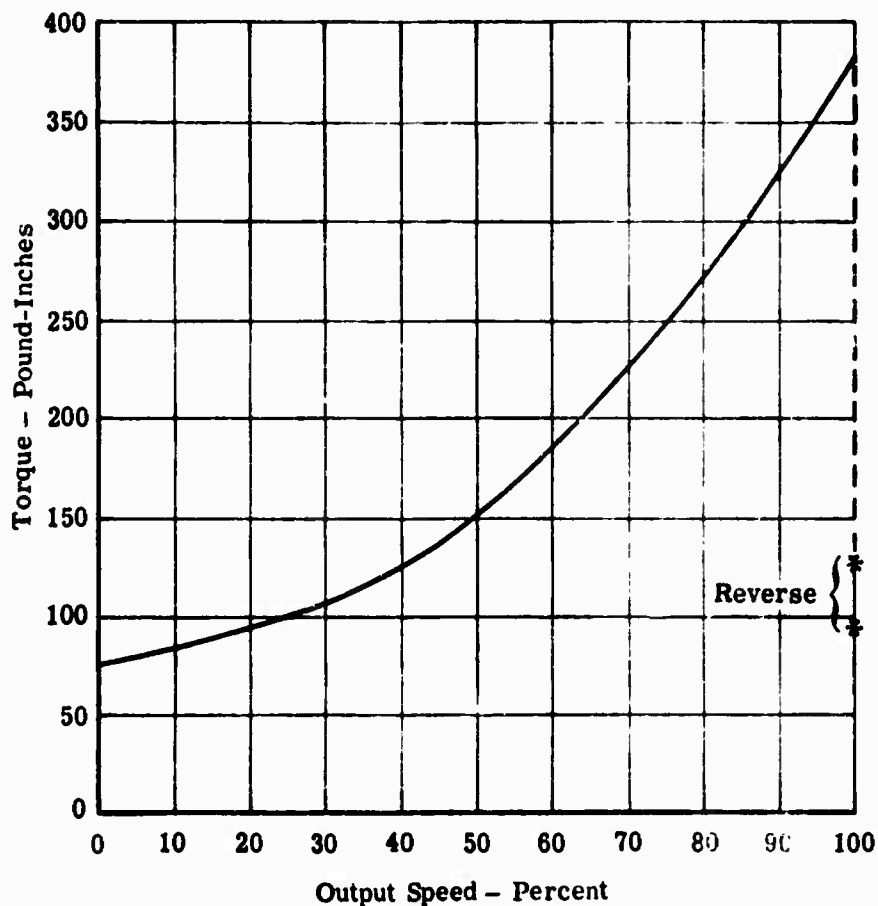
Figure 103. Torque Characteristics—Cone Synchronizing Clutch

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Except for operation of one internal valve which controls engage or disengage, all sequencing in the clutch assembly, including the control of cooling oil, is accomplished automatically. This simplifies the pilot control to a minimum and eliminates the danger of clutch damage due to pilot error. The main clutch has actuating pistons on each end of the plate pack to achieve more uniform pressure on all the plates.

Referring to Figure 102, the upper half of the assembly is shown in the normal disengage position and the lower half in the engage slipping position. A description of the operation of the assembly follows. With the engine operating, the cone outer member "A" rotates with the engine, and oil pressure is transmitted to the assembly at "B." The valve sleeve "C" is moved to the right to admit oil pressure to engage chamber "D." This causes the cone clutch to engage and starts acceleration of the main clutch outer member. The acceleration inertia torque acting on blocker splines "E" prevents the engagement of teeth "F" until speeds are synchronized and the acceleration torque is relieved. The applied and centrifugal oil pressures in "D" cause the member "G" to move to the left, as shown in the lower half of Figure 102. Porting in chamber "D" is uncovered so that oil pressure is transmitted to inside valve mechanism "H." From here, oil pressure is introduced into the clutch-apply chambers "I" and the valve piston chamber "J." The applied pressure causes the clutch-apply pistons and the valve "H" to move to the position shown in the lower half of the drawing. Cooling oil flow is then admitted to the plates through a system of ports as shown, and also to the disengage spring annuli "L." The inner diameter of these annuli in combination with the Belleville springs is sized in relation to chambers "I" to give the desired slip-torque characteristics as a function of output speed. Chamber "K" is kept full of oil under insignificant applied pressure by a system of orifices. As the clutch inner member approaches 90 percent rated speed, the centrifugal head pressure in "K," plus the spring force, overcomes the applied pressure and centrifugal head pressure in "J." The valve "H" then returns to the position as shown in the upper half of Figure 102. The high cooling oil flo and the oil supply to "L" are shut off. The oil in "L" then drains out through bleed holes and the full centrifugal head pressure in "I" greatly increases the force on the clutch plates. The clutch is now able to transmit full engine torque, with substantial margin, without slip.

To disengage normally, valve "C" is returned to the position as shown in the upper half of Figure 102 with oil pressure admitted to chamber "M." This force plus springs "N" causes member "C" to move to the disengage position. The main clutch stays engaged due to centrifugal

head pressure in chamber "I" as long as the rotor speed is high—either being driven by other engines or autorotating. Under this condition, the engine can be restarted and valve "C" moved to the engage position. Member "G" will automatically reengage when the engine reaches synchronous speed. When the rotor slows to a low speed at shutdown, the springs in "L" overcome the centrifugal pressure in "I" and the clutch pistons are moved to the disengage position.

As mentioned previously, the cone clutch assembly also serves as an automatic disconnect device or overrunning clutch under reverse torque. Ramps on the reverse side of teeth "F" force member "G" to the right or out of engagement. Member "O" follows part of the way by a camming action of helical splines "P." This is to ensure that member "G" moves far enough so that blocker spline "E" can engage its detent ramps. Spring "Q" then causes member "O" to return to its normal position. Teeth "F" are prevented from ratcheting by the cone clutch and blocker splines "E." Splines "R" are helical so that they add to the piston force in "D" for normal direction and subtract from it under reverse torque. This reduces the drag on the cone clutch for reverse direction so that it does not overheat even under high-speed reverse slippage. This small drag can then be eliminated by moving valve "C" to disengage; the cone surfaces will then be separated.

SAFETY DEVICES

Safety devices associated with the transmission system will depend to some degree on the type of engine selected for HLH. If fixed shaft engines are used, it will be necessary to incorporate a clutch, which can be controlled from the cockpit, between each engine and the combining gears in the transmission. These clutches must be disengaged while engines are being started. When the engines are at the proper speed, the clutches can be engaged (singly or collectively, depending on engine and rotor characteristics) to bring the rotor up to speed. Engine failures in systems of this type can be extremely serious, since the power from one to two more engines will be absorbed in maintaining the inoperative compressor at system r.p.m. until flight personnel can assess the problem and disengage the proper clutch. Therefore, it is essential that an overrunning clutch be arranged in series with each starting clutch so that individual power units cannot absorb power from the system. (This feature is provided in the compact clutch design previously shown in Figure 102.) If free-turbine engines are used in the HLH, starting clutches are not required because the gas producers would not be mechanically coupled to the rotor. However, gas coupling does exist and

inoperative turbine power absorption is sufficiently high to justify the use of the overrunning clutches. In addition, if free-turbine engines are to be run on the ground for control adjustments, etc., it is desirable to introduce a rotor lock mechanism to prevent the rotor from turning and creating downwash while loading the vehicle or when conducting engine preflight tests. The service life of current helicopter engines of all types is reduced by dust and other debris introduced into engine air inlets by rotor downwash not only during takeoff and landing, but also during ground run-up tests which constitute a high percentage of total military engine time.

Unfortunately, it appears that the rotor lock cannot be completely integrated with a rotor brake due to large differences in torque requirements. The stall torque of a free power turbine with its gas producer operating at idle speed (for electric power, control adjustments, or instant takeoff) will be as much as 65 percent of rated torque. With the assumption that a 15,000-horsepower transmission will be required for the HLH, it is estimated that the rotor lock should be designed with the capability to withstand at least a 4,000,000-pound-inch torque at rotor speed. In addition, the lock should be designed to disengage while subjected to design torque.

The rotor brake must be designed to meet considerably different requirements. To avoid rotor damage caused by striking other equipment in congested loading areas, it is desirable that rotors be braked to a stop as soon as possible after landing and be kept from turning under the influence of prevailing winds. On the other hand, rotor deceleration should not be so fast that blades are damaged by inertia loads. Specific rotor data have not been made available to Allison, but it is believed that the following design parameters are applicable. For rotors turning at 150 r.p.m. with a polar moment of inertia of approximately 130,000 slug-feet squared, the brake should be designed to stop the system in approximately 30 seconds. Brakes of the type recommended by Allison, based on experience with Model T56 and T78 engines—brakes which provide increasing torque along a squared curve from zero at the engaging speed to a maximum value at stall—should be designed for a stall torque of 220,000 pound-inches (expressed at the rotor shaft). Brake life is extended by applying low torques at the higher slip speeds to reduce heat rejection and lower brake surface temperatures. Increased brake torques at low slip speeds are most beneficial because this is the region in which windage is no longer significant in stopping the rotor. Unlike the rotor lock which is recommended only with free turbine engines, the brake should be incorporated regardless of the type of engine which is ultimately selected for the HLH.

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Design features of the clutch for fixed shaft engines were discussed previously. Design features of the remaining safety devices are discussed in the following paragraphs.

Overrunning Clutch

Based on turboprop engine experience, disk-type overrunning clutches were selected over sprag or roller types for both the gas- and gear-coupled transmissions in this power category. Sprags tend to create large shock loads on gears, bearings, and shafts when power turbines are accelerated into the rotor system with values which easily exceed 10,000 r.p.m. per second. Sprags also tend to move over center and jam. Rollers are susceptible to slipping when subjected to torsional shaft vibration, require excessive manufacturing control of cam angles, ar ! create high Hertzian stresses. Disk clutches provide adequate slippage during high-speed engagements to reduce impact loads and yet are capable of transmitting design torques without slipping. Internal stresses are relatively low and the design is insensitive to torsionals. Plate warping has not proved to be a problem, since no significant power is transmitted when overrunning. In this design, clutch disks are kept in intimate contact while overrunning (by a combination of springs and centrifugal oil pressure) to provide immediate engagements when the torque path is driving the rotor. The cavity employed to create centrifugal oil pressure is located on the turbine or input shaft so that clamping force on the clutch disks when overrunning is reduced as the turbine is shut down and coasts to a stop.

Tentative overrunning clutch design parameters are as follows:

Design Torque	16,300 pound-inches
Number Of Friction Surfaces	10
Friction Surface:	
Outside Diameter	4.875 inches
Inside Diameter	3.750 inches
Unit Area	7.6 square inches
Outer Plate Material	Steel
Inner Plate Material	Steel with sintered bronze bearing
Spline Helix Angle	40 degrees
Spline Pitch Diameter	2.625 inches
Estimated Weight	20 pounds
Oil Flow Rate	12 pounds per minute

No external controls are required, since clutch operation is automatic. A representative overrunning clutch design is shown in Figure 91.

Rotor Lock and Brake Assembly

A rotor lock and brake assembly driven at 19,300 r.p.m. by the main combining gear was studied for the HLH. One design is applicable to either the gear- or gas-coupled transmission, although differences are required in support bearing and shaft geometries to suit the different installations. By placing the lock and brake assembly on high-speed shafts, torque forces are reduced, permitting a smaller, lighter design. To reduce maintenance time and cost in the field, the assembly can be removed for inspection or replacement without disassembling the transmission. The spline lock is sized to hold the rotor stationary with all engines running at idle speed and is provided with an actuator which is capable of exceeding spline reaction force at maximum power and disengaging the lock while engines are maintained at this setting. Sufficient margin is provided in the lock design stresses so that failure will not occur in the event that engines are inadvertently operated at maximum power; it is not considered necessary or desirable that locked rotor operation be conducted at this torque setting. Inadvertent attempts to engage the lock while the rotor is turning are made harmless by a hydraulically actuated blocker valve. Transmission oil pressure which will be present when the rotor is turning will cause the valve to interrupt the air supply to the lock actuator and make it inoperative. For normal locked rotor starts, starter supply air would be diverted to the lock actuator at a minimum pressure of 20 p.s.i.g. As the piston moves to engage the lock, the internal lock piece will rotate until the spline teeth are aligned and engaged. The splines would incorporate a helix angle arranged to oppose the actuator with a force of 2700 pounds when subjected to a design torque of 31,000 pound-inches at the lock. To obtain sufficient actuator force at the higher lock torques, compressor discharge air is manifolded through one-way valves into the starter air supply line, as shown in Figure 93. Thus, the lock separating forces which vary with engine torque are opposed by compressor discharge air pressure, which creates, and is proportional to, turbine torque. To disengage the lock, the actuator air supply line is vented to the atmosphere so that spline tooth torque reaction force will cause the teeth to separate. The lock is held disengaged by a light spring.

The rotor brake is patterned from the design being developed for Model T78 engines. For normal starts with the rotor unlocked, starter-engine air at a minimum pressure of 20 p.s.i.g. is applied to a brake actuator piston. The air pressure creates a torque on the outer cone brake member through rack teeth cut on the actuator piston to engage teeth on the outside of the outer brake cone.

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This torque is sufficient to cause the cone to rotate on helical splines and move axially against a Belleville spring and out of engagement with the inner brake cone. As the transmission speed increases, oil is supplied to a rotating chamber in the inner cone where it creates a centrifugal head. At 30 percent of rated speed, this oil head is adequate to move the inner cone against a second spring and increase the clearance from the outer cone. Above this speed, interruptions in air pressure will not engage the brake because of inner cone motion. Interruptions in the transmission oil supply will not cause the brake to engage because it is not dependent on pump pressure. During rotor coast down, a pressure-sensitive valve diverts transmission oil to the brake cones for cooling at 40-percent speed. At 30-percent speed, the inner cone is moved into contact with the outer cone by a Belleville spring opposing the centrifugal oil head. As oil pressure continues to decrease with speed, brake torque increases from zero at engagement to a maximum value of 1700 pound-inches (at the brake) when stationary.

The estimated weight for the rotor brake and lock assembly with internal valves and lines is 40 pounds. External air lines and valves are not included, due to lack of information relating to a specific installation.

GAS-COUPLED ROTOR DRIVE

Installation of the power turbine(s) in a vertical position on the transmission was studied as a method of eliminating the problems associated with high power transmission through right-angle gears. This arrangement allows a high degree of flexibility in the location of the gas producers. Since it is unnecessary to consider gear geometry when locating engines, it is possible to arrange the transmissions for best rotor performance and to locate the gas producers to suit airframe profiles. Preliminary design studies were conducted on multi- and single-turbine versions of the gas-coupled arrangement.

Multiturbine Arrangement

The most straightforward evolution from contemporary free turbine engines to the remote gas-coupled arrangement is to remove the power turbine from each engine and relocate them on the transmission in the vertical position. By providing a separate power turbine for each gas producer, it is unnecessary to introduce hot gas valves into the system. Multiple turbines provide high efficiency at reduced power levels with one or more gas producers inoperative, since operating temperatures and gas flows can be held near optimum values, while admission and

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filling losses are avoided and large wheel temperature gradients are eliminated. System reliability is improved because of redundant power paths, and system weight is reduced by high wheel speeds and low torques.

Figure 104 was prepared to assist in determining scaling factors such as turbine duct clearances and preliminary weight estimates. Design parameters for the single-rotor configuration are as follows:

	Three Engines	Four Engines
Overall Gear Ratio	128:1	128:1
Number Of Turbine Drives	3	4
Turbine Speed—r. p. m.	19,320	19,320
Cross Shaft Speed—r. p. m.	19,320	19,320
Rotor Speed—r. p. m.	150	150
Continuous Rotor Moment—pound-inches	1,500,000	1,500,000
Rotor Torque—pound-inches (85-percent power)	5,350,000	7,120,000
Total Horsepower (Rotor + Cross-Shaft)	15,000	18,000
Weight Summary—pounds		
Transmission (Estimated)	3,380	3,550
Turbine Rotor	155	205
Duct and Structure	450	600
Gas Producer	1,665	2,220
Total	5,700	6,575

Preliminary gear data established for the configuration are presented in Table 13.

Turbine seals are arranged to provide a wheel force balance using compressor discharge air beneath the turbine rotors. While this problem is somewhat unique in transmission design, it is no more difficult than balancing on an engine. The basic transmission would use clutches, planetary gears, rotor brake, and rotor lock geometries which are unchanged from the configurations studied for use with bevel gear drives. However, it is recommended that a double helical combining gear system be used with the gas-coupled design. Double helicals can be machined more accurately because of the narrower face widths of individual gear rings; also they offer a stabilizing tooth action which is particularly desirable when large gear ratios and center distances are required. In

this particular design. turbine and gas duct clearances are primarily responsible for the 18-inch center distance between input pinions and the combining gear. By using double helicals in the first gear stage, it is believed that the turbine clearance can be increased as much as 30 percent without resorting to idler gears or encountering excessive pitch-line velocities. Many of the features studied for single rotor transmissions are directly applicable to dual units. Primary differences between the two concepts are reflected in the cross-shaft drives and the planetary output gear stage, due to a different power distribution. Cross shafting for single rotor units must transmit from 15 to 25 percent of the total rated power, with the remainder passing to the rotor. Dual systems will be affected to a greater degree in this area by the engine arrangement finally selected for the HLH. With all engines in the rear of the vehicle, the shafting must transmit up to 60 percent of the rated power. If four power units of equal size are installed with two at each end of the vehicle, the cross-shaft can be designed to transmit 30 percent of the rated power. Without considering the weight and balance or packaging problems associated with a specific vehicle, it can be definitely stated that from the viewpoint of a transmission designer it is preferable to mount power units at each end of a dual rotor vehicle so that cross-shaft power can be reduced (in this case from as much as 9000 horsepower). By resorting to a supercritical design, the weight and complexity of the shafting can be improved, but a serious problem will still exist in the fact that 9000 horsepower must be transmitted through at least one and perhaps two spiral bevel gear meshes.

TABLE 13
ESTABLISHED PRELIMINARY GEAR DATA

Gear	Number Required	Number of Teeth	Normal Pitch (in.)	Pitch Diameter (in.)	r.p.m.	Tangential Tooth Load (lb.)	Face Width (in.)
Pinion	3	26	8	3.75	19,320	8,700	2 × 1.5
Combining Gear	1	224	8	32.4	2,250	8,700	2 × 1.45
First Sun	1	27	3	9.0	2,250	19,800	4.0
First Planet	4	53	3	17.66	377	19,800	4.0
First Ring	1	133	3	44.33	-274	19,800	3.0
Second Sun	1	91	4	24.75	-274	17,700	3.9
Second Planet	8	41	4	10.25	663	17,700	3.9
Second Ring	1	181	4	45.25	150	17,700	2.9

Section A-A

Design Input Power   = 1
Turbine r.p.m.   = 1
Rotor r.p.m.   = 1
Cross-Shaft r.p.m.   = 1
Cross-Shaft Power   = 2
Estimated Weight   = 4

Figure 104. Gas-Coupled Dri

181

Scavenge Oil Outlet Pad
Similar to ARP 454A

Pressure Pump Inlet Pad
Similar to ARP 454A

Alternator Pad
Type XVI-B Modified Per AS470

Alternator Envelope

℄

Cross shaft

Cross shaft

A

Center of Turbine

Center of
Turbine

Center of Turbine

A

View Looking Up
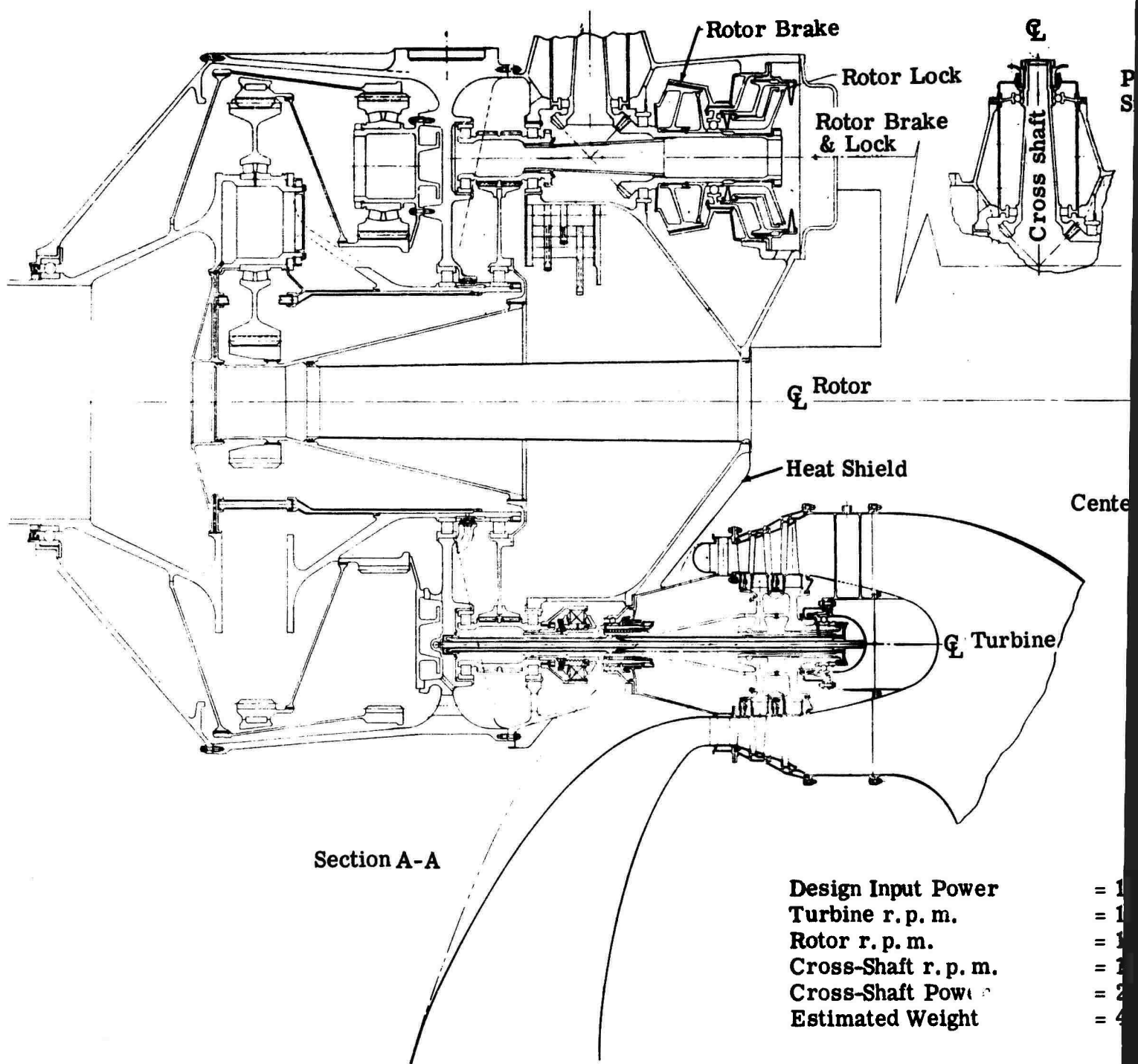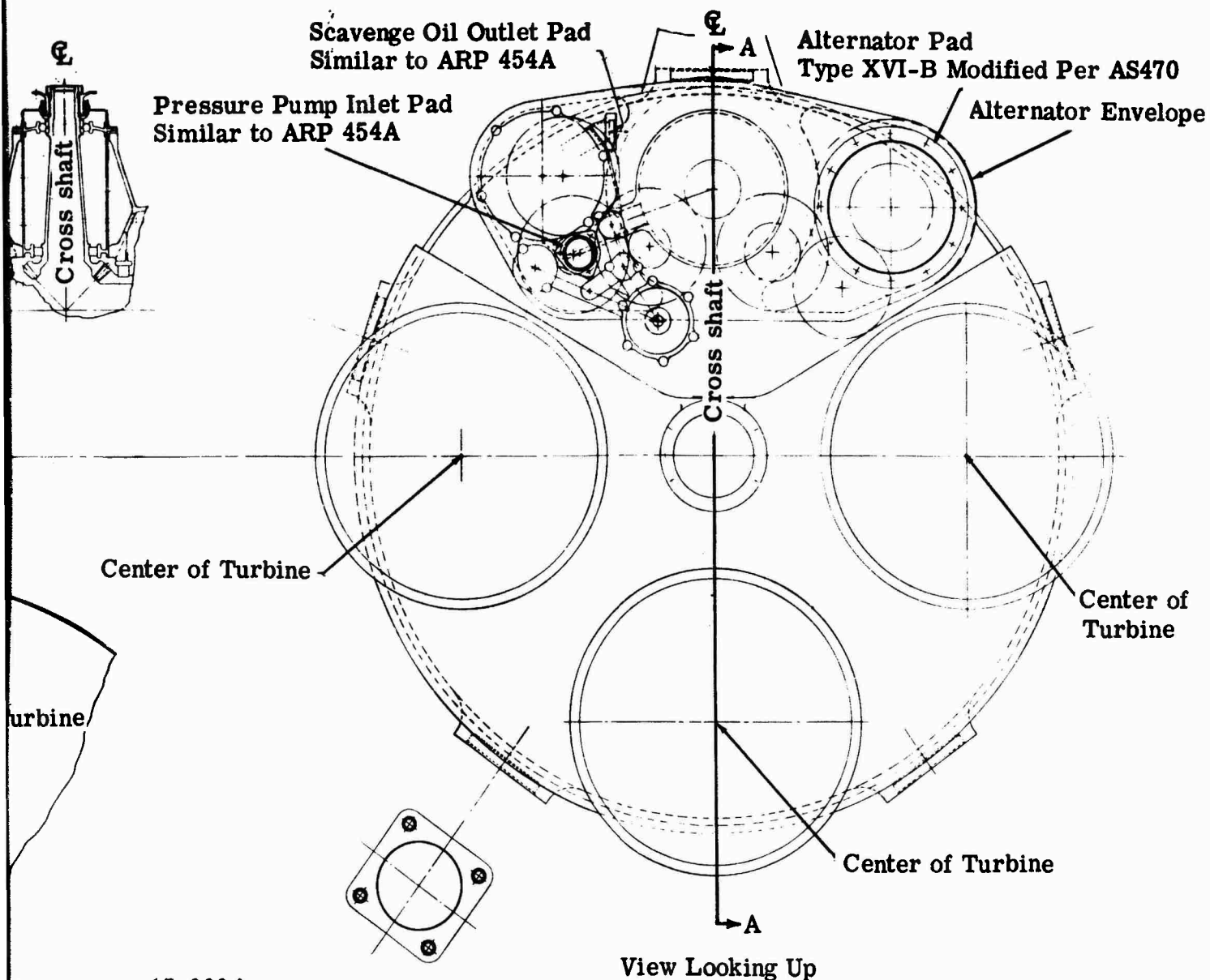
urbine

r   = 15,000 hp
            = 19,300
            = 150
            = 19,300
            = 2,500 hp
            = 4,000 pounds

Coupled Drive — Multiturbine

2

The following estimated design parameters have been established for a dual-rotor transmission for comparison with the single-rotor unit requirements already described.

Overall Gear Ratio	128:1
*Number of Turbine Drives	1 or 2
Turbine Speed	19,320 r. p. m.
Cross-Shaft Speed	19,320 r. p. m.
Rotor Speed	150 r. p. m.
Total Horsepower	9000
*Estimated Transmission Weight	2300 pounds
*Turbine Rotor Weight	77 pounds
*Duct and Turbine Structure Weight	225 pounds
*Total Weight of One Transmission	2602 pounds
(Two required per vehicle)	

The transmission illustrated by Figure 104 has been arranged so that all oil pumps, valves, and the rotor lock and brake assembly can be removed for inspection without disturbing the transmission. Turboprop engine experience has shown that these features are essential in reducing maintenance and in obtaining maximum utilization of the vehicle. Estimated oil flow for the design is 71 gallons per minute, based on an overall transmission efficiency of 98 percent. This compares favorably with right-angle gear designs (not gas coupled), where it is estimated that higher gear losses will require an oil flow of approximately 105 gallons per minute. Lower oil flows in the gas-coupled transmission will permit the use of smaller oil coolers, supply tanks, pumps, valves, and oil lines.

At the beginning of this study, it was anticipated that gas-coupled transmissions might incur a serious weight penalty caused by the insulation required to prevent radiant heat transfer between gas ducts and transmission castings. This did not prove to be a problem. Heat transfer analyses performed at Allison have shown that radiant flux density into the transmission is negligible, with no heat shield provided. Transmission wall temperatures would remain below 250°F. and oil cooler sizes could remain unchanged. In the event that hot spots are encountered

---

*With the assumption that three engines will be capable of meeting HLH requirements, it is believed that two essentially identical transmissions can be used. The weights were estimated by prorating 1 1/2 turbine drives to each transmission.

due to specific duct-transmission geometries which were not considered, it would be feasible to provide a forced draft from the rotor downwash. The heat input from one duct to the adjacent housing was calculated to be 300 B. t. u. per minute based on direct radiation between parallel infinite block plates. Convective heat transfer by air flowing past the transmission would serve as a coolant. For flight safety, it is believed that a lightweight heat shield will be required to protect the transmission castings, control components, and oil lines from high temperature gas leaks out of the duct system.

The pressure losses associated with the ducting and diffusers of this design can be divided into two sections:

- The region between the gas-producer turbine and the power turbine
- The exhaust duct or the region downstream of the power turbine

In an effort to keep the overall pressure loss as low as possible, a diffuser should be incorporated following the gas-producer turbine. The diffuser assumed for this study resulted in a reduction in the Mach number from approximately 0.40 to 0.17. This results in small losses due to friction and turning in the ducting and the turbine inlet scroll. The overall losses in this region are expected to be 4 percent of the total pressure at the discharge of the gas-producer turbine.

The exhaust ducting consists of a diffuser with a 2:1 area ratio and a constant area 90-degree turn followed by a diffusing outlet duct. In an effort to keep the diffuser length reasonable and minimize weight, one splitter vane was utilized. The length of the diffuser is approximately 13 inches. The performance of the exhaust ducting was incorporated in the performance estimate of the power turbine.

No significant difference can be found in the system weights of geared or gas-coupled transmissions if it is assumed that the power turbine weights are transferred from the gas producer to the transmission. The gas-coupled approach eliminates power shafts and thrust bearings from the first gear stage but will require additional turbine support structure and slightly longer gas ducts. The obvious advantage of the gas-coupled system lies in the elimination of high-power bevel gear drives and in installation flexibility.

Five gas producer arrangements were studied to investigate mechanical limitations which might be encountered in a gas-coupled HLH design. The five arrangements are as follows:

184

- Two T78 gas producers plus reheat
- Three T78 gas producers plus water injection
- Four T78 gas producers
- Four T78 gas producers plus regeneration
- Four T78 gas producers plus wings and auxiliary propellers (compound helicopter)
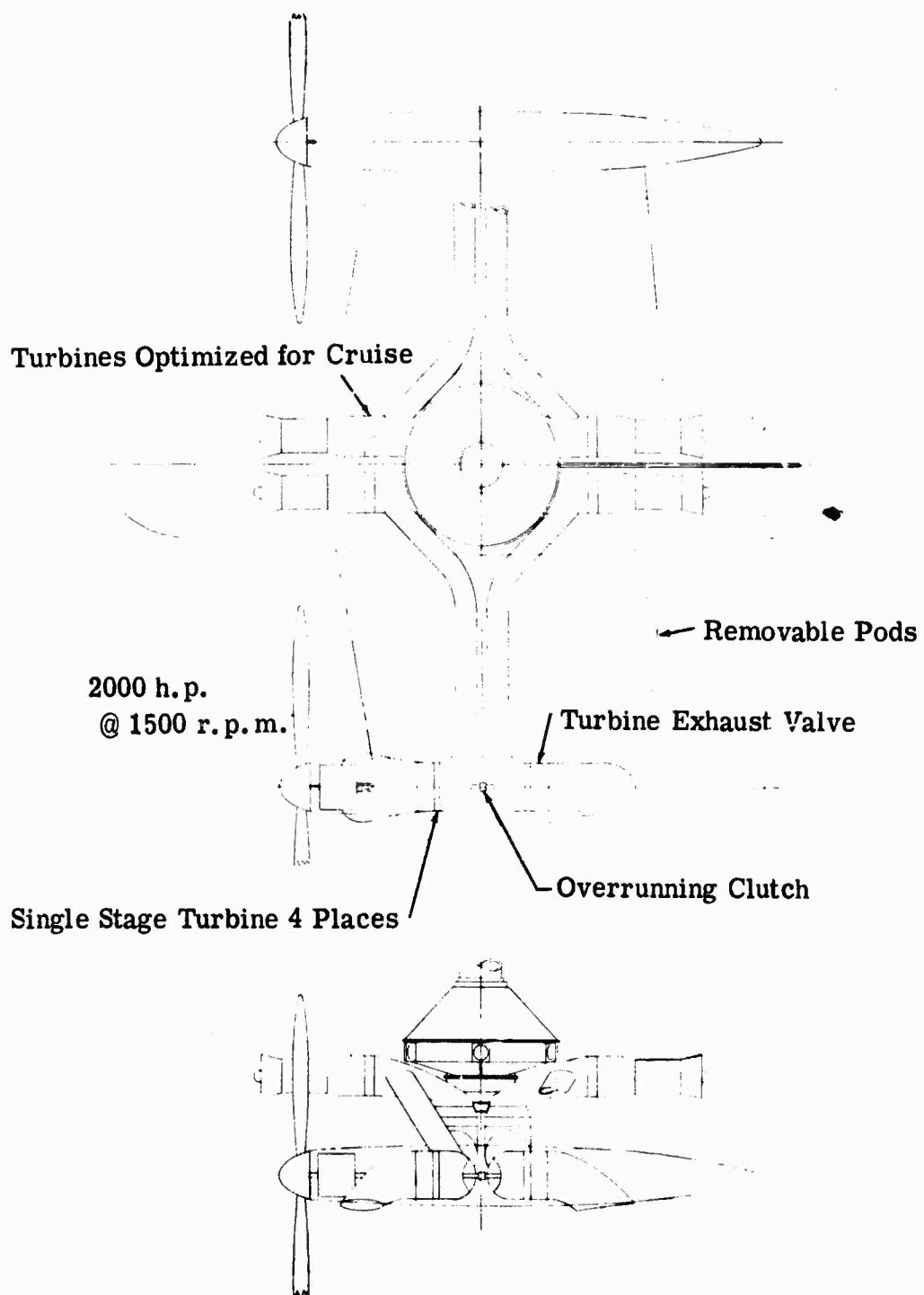
Since it is probable that compound helicopters will be capable of achieving the higher airspeeds which minimize the effects of adverse winds and extend range on ferry missions, the compound arrangement was included to establish the applicability of gas-coupled systems to compound machines. The approach to a compound helicopter drive shown in Figure 105 features a simplicity which permits rapid installation and removal of wings and auxiliary propulsion units by semiskilled personnel. It is realized that evaluation of this approach would require a detailed vehicle-mission analysis which is not within the scope of this study. It is shown to indicate the flexibility of the gas-coupled approach.

Single- and dual-rotor applications were considered for each arrangement, and it appears that gas coupling does not introduce fundamental mechanical limitations to any of the systems studied. In fact, the gas coupling approach can offer another degree of freedom for the designer when fitting a propulsion system into a specific aircraft by allowing arrangement of engine positions and angles without restriction to practical gear geometries.
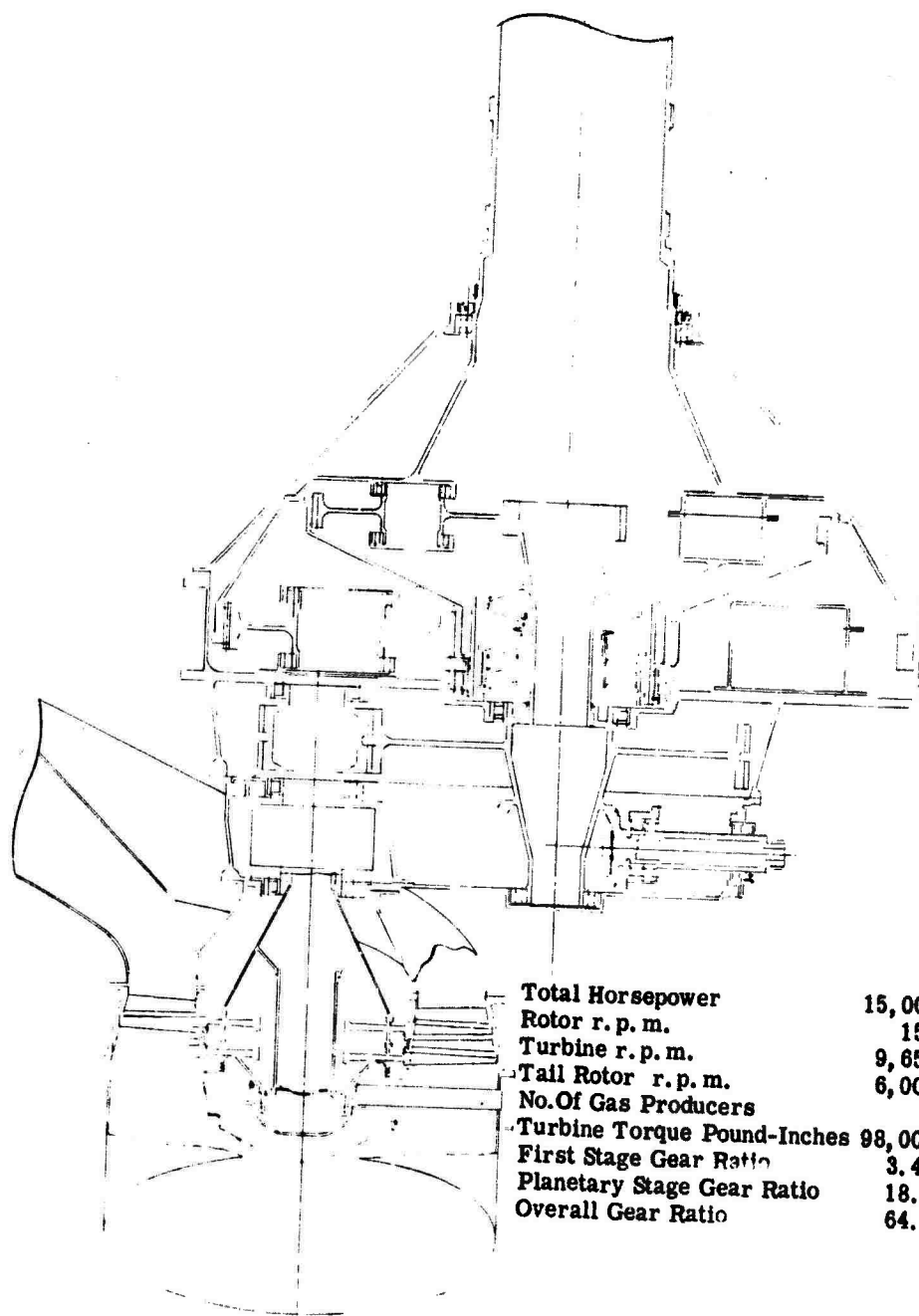
Single-Turbine Arrangement

A single-turbine drive (Figure 106) was studied for comparison with a four-turbine unit arrangement. It was assumed that four gas producers would feed quadrants of the turbine inlet annulus with a common exhaust duct. Tip speed was held constant to maintain high work capability per turbine stage. This resulted in a single 36-inch-diameter, two-stage turbine turning 9650 r.p.m. and delivering 15,000 horsepower versus four 18-inch-diameter turbines which can deliver the same power at 19,300 r.p.m. Static exhaust pressures were established to provide favorable airflow through inoperative gas producers without requiring valves to prevent backflows. A one-way clutch was included to eliminate turbine drag during autorotation.

The single-turbine system proved attractive to the extent that fewer gears and bearings are required. Conversely, the single turbine is heavier, less reliable, and less efficient at part power than the multiple

185

Turbines Optimized for Cruise

2000 h.p.
@ 1500 r.p.m.

Removable Pods

Turbine Exhaust Valve

Overrunning Clutch

Single Stage Turbine 4 Places

Figure 105. Gas-Coupled Compound Helicopter Drive

186

Total Horsepower	15,000
Rotor r.p.m.	150
Turbine r.p.m.	9,650
Tail Rotor r.p.m.	6,000
No. Of Gas Producers	4
Turbine Torque Pound-Inches	98,000
First Stage Gear Ratio	3.40
Planetary Stage Gear Ratio	18.9
Overall Gear Ratio	64.4

Figure 106. Single-Turbine Drive

187

turbine design. Turbine torque was increased eight times due to the constant tip speed limitation so that clutch gear and bearing loads increased in proportion and produced an unfavorable weight trade-off. System reliability was reduced by the elimination of four parallel power paths to the combining gear in trade for a single path. Reliability may be further compromised by thermal stresses in the turbine blades resulting from partial admission at reduced power settings. Efficiency of the single-turbine design is reduced by admission and filling losses with one or more gas producers inoperative and by reduced inlet temperatures or wheel speeds which may be needed to compensate for thermal stresses resulting from partial admission.

It was concluded that multiple-high-speed turbines are lighter, more efficient, and more reliable from an overall system viewpoint than single-power turbines.

(C) INSTALLATION STUDIES (U)

(U) OUTLINE DRAWINGS

(U) Outline drawings for the engine models used in this study (Models 545-C2, 545-C3, 546-C2, 546-C3, 548-C2, 501-M25, and 501-M26) are shown in Figures 107 through 111. It should be noted that the envelopes for these engines are similar and fall within a narrow range of dimensions. It can be assumed that these engines are interchangeable for the purpose of conducting preliminary installation design studies.

(U) INSTALLATION DATA

(U) A summary of engine characteristics and installation data is presented in Table 14.

(U) Historically, helicopter engine oil cooling has been provided by an oil-to-air heat exchanger. If a fuel-to-oil heat exchanger could provide adequate cooling, weight reduction and system simplification may be realized. Therefore, a study was made to determine the feasibility of using engine fuel to cool the engine oil.

(U) The results of this study indicated that adequate oil cooling could be accomplished in this manner for all flight conditions. However, under certain ground idle conditions, an operating time limit may be required. For example, the worst condition for the 501-M25 was found to be high-speed ground idle (13,200 r.p.m.). The hot-day heat rejection and fuel flow for this condition are estimated to be 1000 B.t.u. per minute and 10.4 pounds per minute, respectively. With a maximum oil inlet temperature to the heat exchanger of 270°F., a fuel inlet temperature of 100°F., and a heat exchange efficiency* of 100 percent, 884 B.t.u. per minute would be transferred to the fuel, thus indicating the fuel to be an inadequate heat sink for continuous operation at this condition.

(U) It is believed that use of low speed ground idle (10,000 r.p.m.) would be more representative of typical helicopter operating procedure. The heat rejection and fuel flow for the 501-M25 at this condition are estimated to be 500 B.t.u. per minute and 8.3 pounds per minute,

---

(U) *Heat exchanger efficiency is 100 percent when maximum temperature of the coolant equals the maximum temperature of the fluid being cooled.

respectively. Using the assumptions of the preceding paragraph, the ideal heat sink capacity of the fuel would be 704 B.t.u. per minute. Thus, indicating a heat exchanger with an efficiency of 71 percent would provide adequate cooling.

(U) In both cases considered, the fuel heat sink capacity is from inadequate to marginal. This implies that a time limit must be placed on ground idle operation. This limit will be dependent on initial temperature of fuel and oil, quantity of oil in the system, maximum allowable oil temperature, and heat exchanger characteristics.

(U) Further consideration of the use of engine fuel to cool the engine oil appears warranted when representative HLH operating procedures and load schedules have been established.

(U) INSTALLATION ARRANGEMENTS

(U) A number of possible engine arrangements for two basic HLH airframe types (single rotor and tandem rotor) were considered. In all of the examples shown, effort was directed to optimize accessibility, provide good engine inlet air conditions, minimize exhaust gas ingestion in inlet air, and achieve a maximum of commonality of power coupling and transmission components.

(U) HLH Single-Rotor, Three-Engine Mechanical Drive

(U) Figure 112 shows a mechanical drive in which three engines are arranged in a common horizontal plane about the rotor transmission. Antitorque rotor shafting is external to the aircraft structure and is covered with a lightweight fairing. Exhaust ducting is arranged in such a manner as to permit rapid engine change and minimize exhaust gas ingestion. Close proximity of the engines to the rotor center line should minimize recirculation of the exhaust and rotor downwash effects. Forward placement of the engines provides a more favorable aircraft center of gravity.

(U) The Models 548-C2 and 501-M26 engines will fit within the envelopes shown in Figure 112. Engine support structure has not been shown due to a lack of detailed airframe definition.
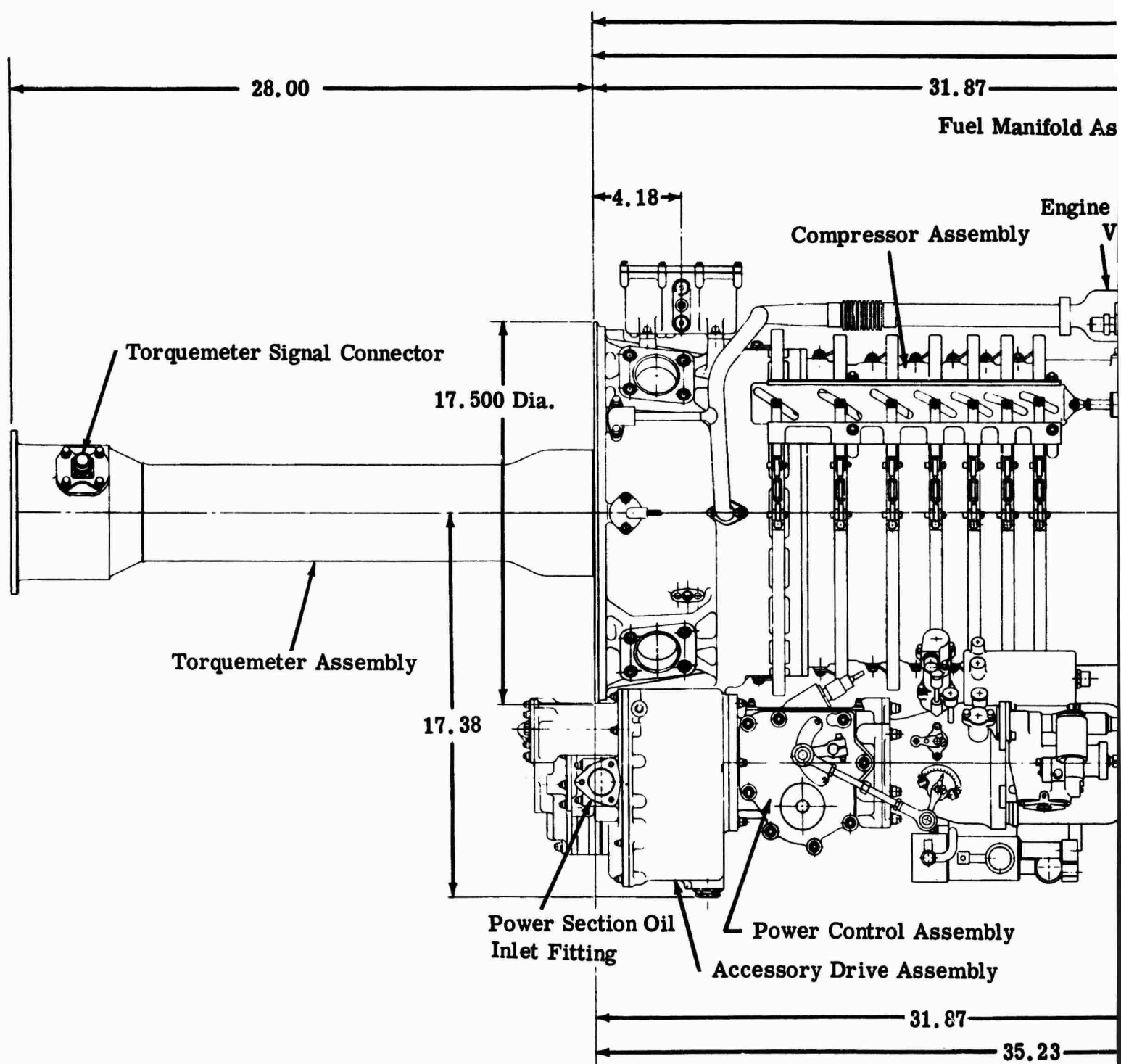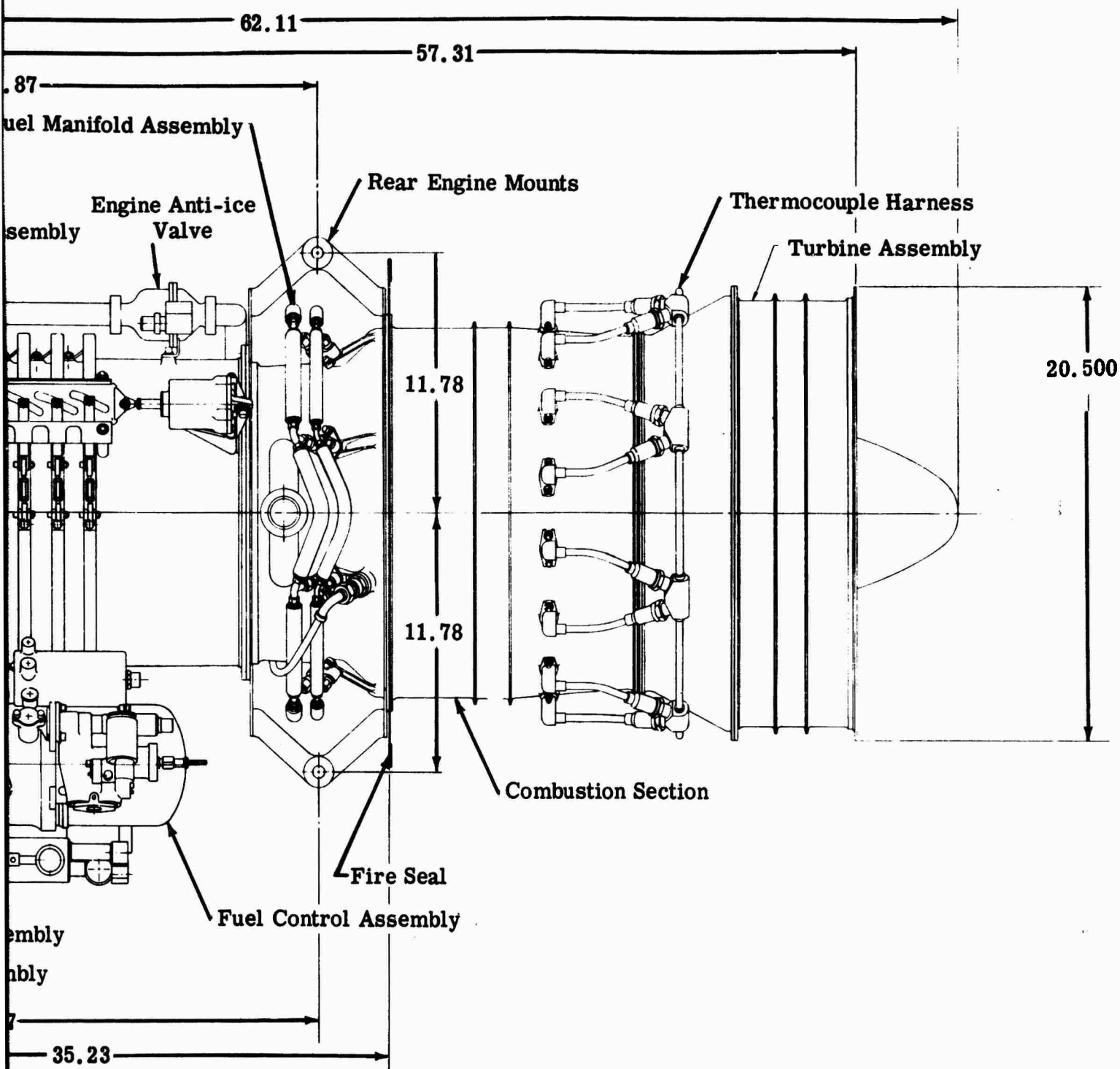
Figure 107. (U) Outline Drawing for Models 54

62.11

57.31

.87

uel Manifold Assembly

Engine Anti-ice
Valve

sembly

Rear Engine Mounts

Thermocouple Harness

Turbine Assembly

20.500

11.78

11.78

Combustion Section

Fire Seal

Fuel Control Assembly

embly

hbly

35.23

g for Models 546-C2 and 546-C3 (SK-14743)

2

Figure 108. (U) Outline Drawing for Models 545-C2

bly

ine Assembly

Regenerator Assembly

18.53

18.53

18.00

36.00
Dia.

17.300
Dia.

Rear Mount (Typ)

Rear View

110.05
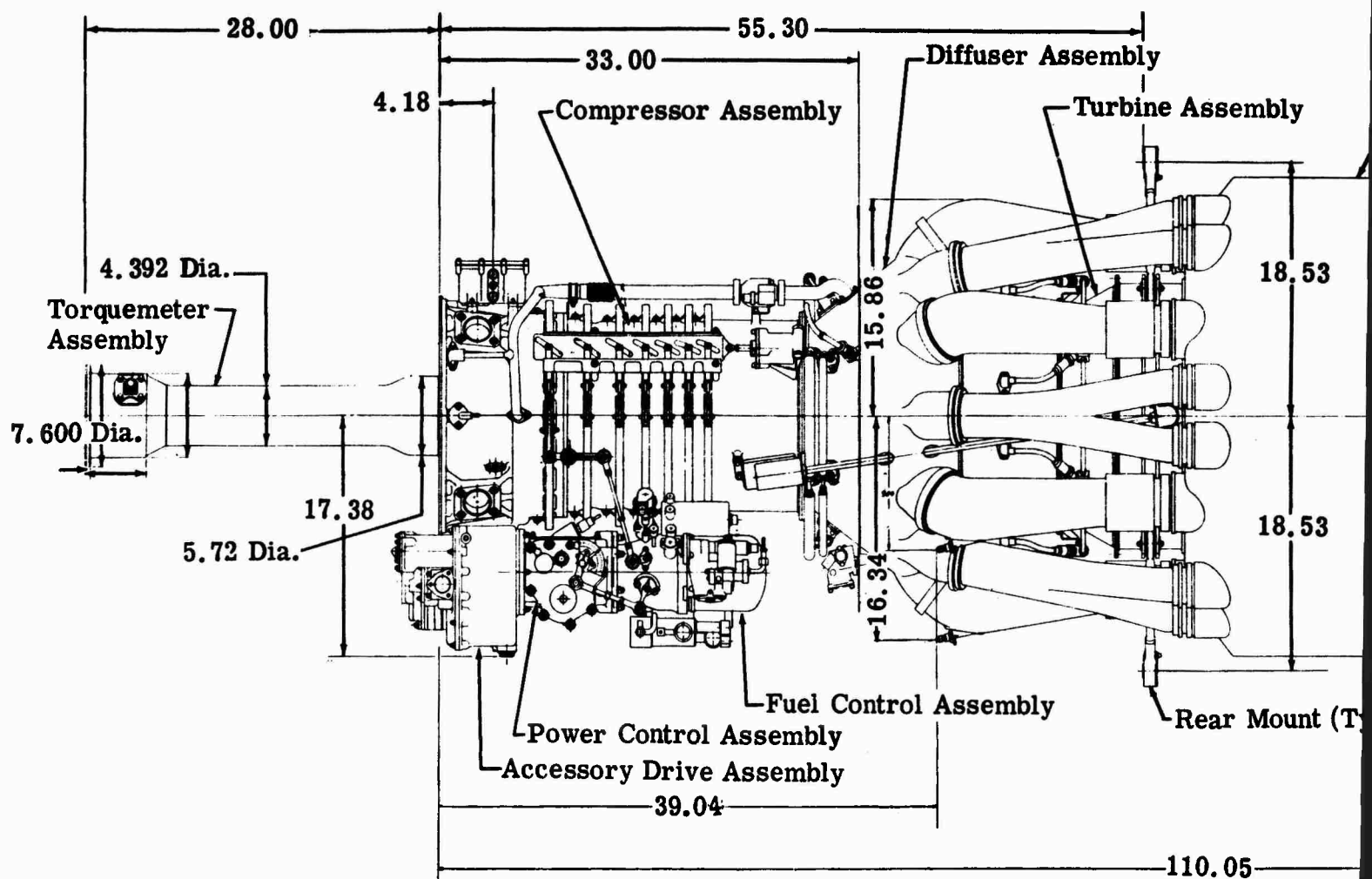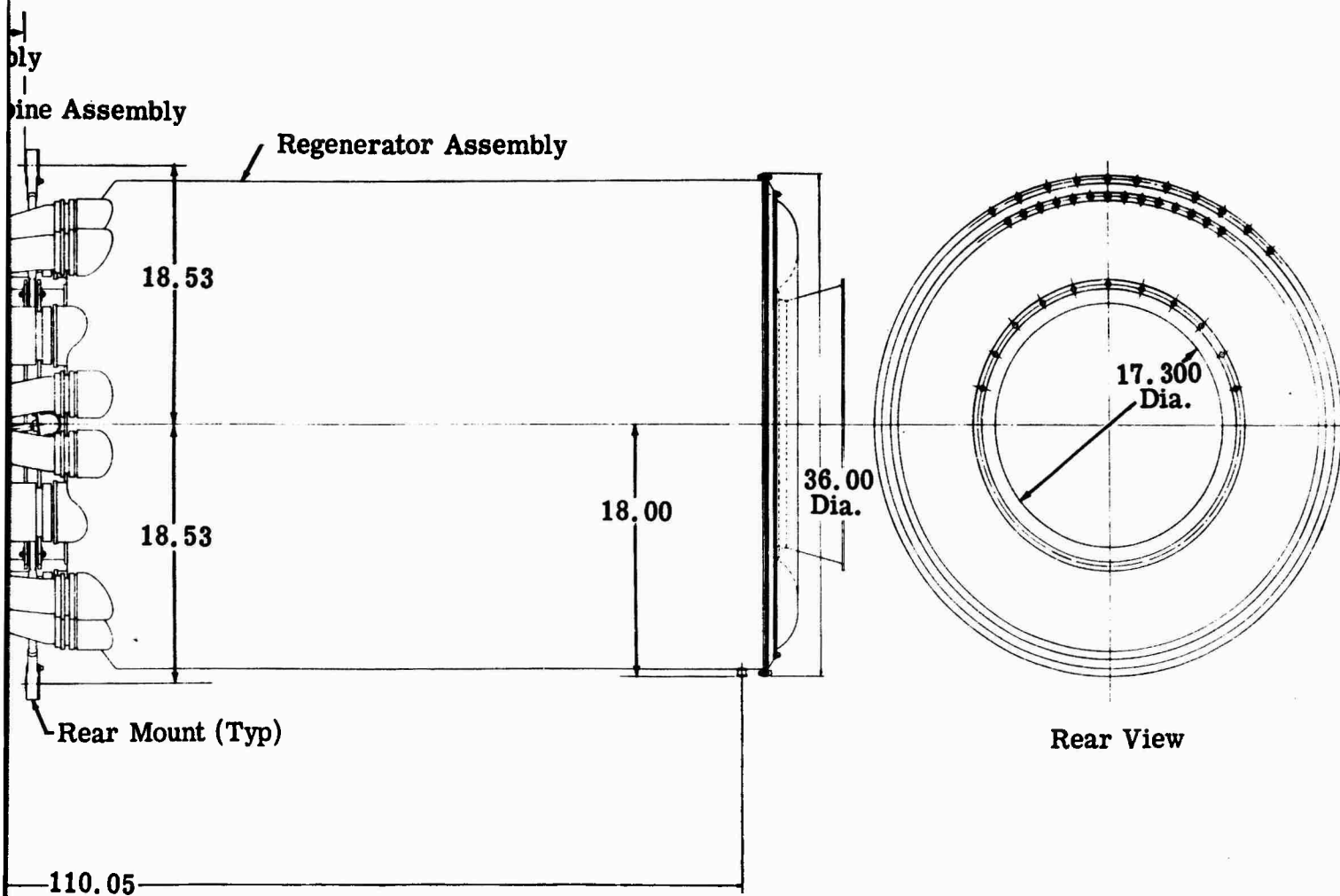
r Models 545-C2 and 545-C3 (SK-14744)

2

Aircraft Furnished
Starter Envelope

4.80

78.07

47.47

34.07

30

Forward Engine
Mount Plane For
Vertical & Lateral Loads

15.50
Dia.

C. G. Of Engine
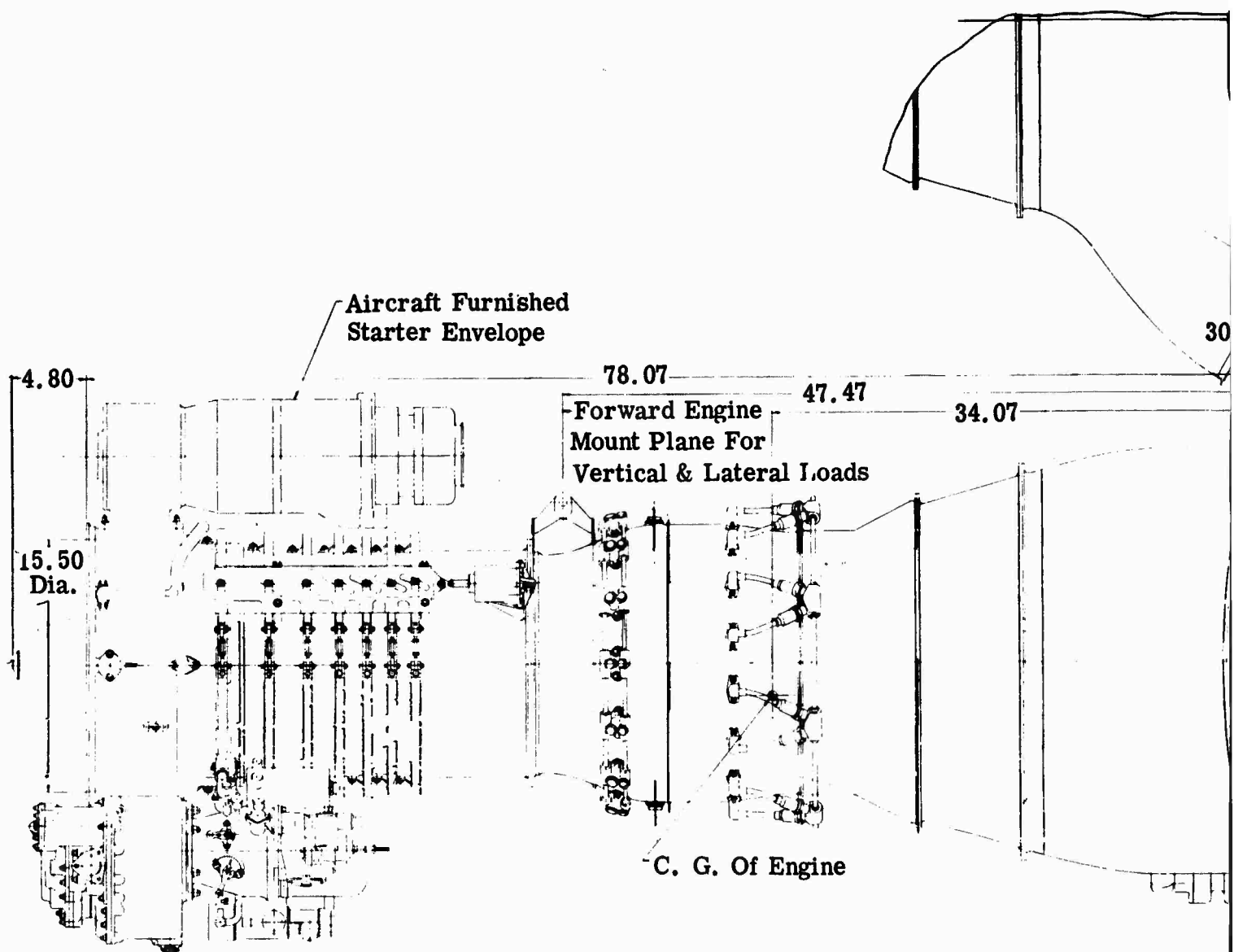
Figure 109. (U) Outline Drawing for Model 54

15.84

14.69

30°

1.48
4.06

Rear Engine Mount Plane
For Vertical, Lateral &
Longitudinal
Loads

13.00

26.00

7.89
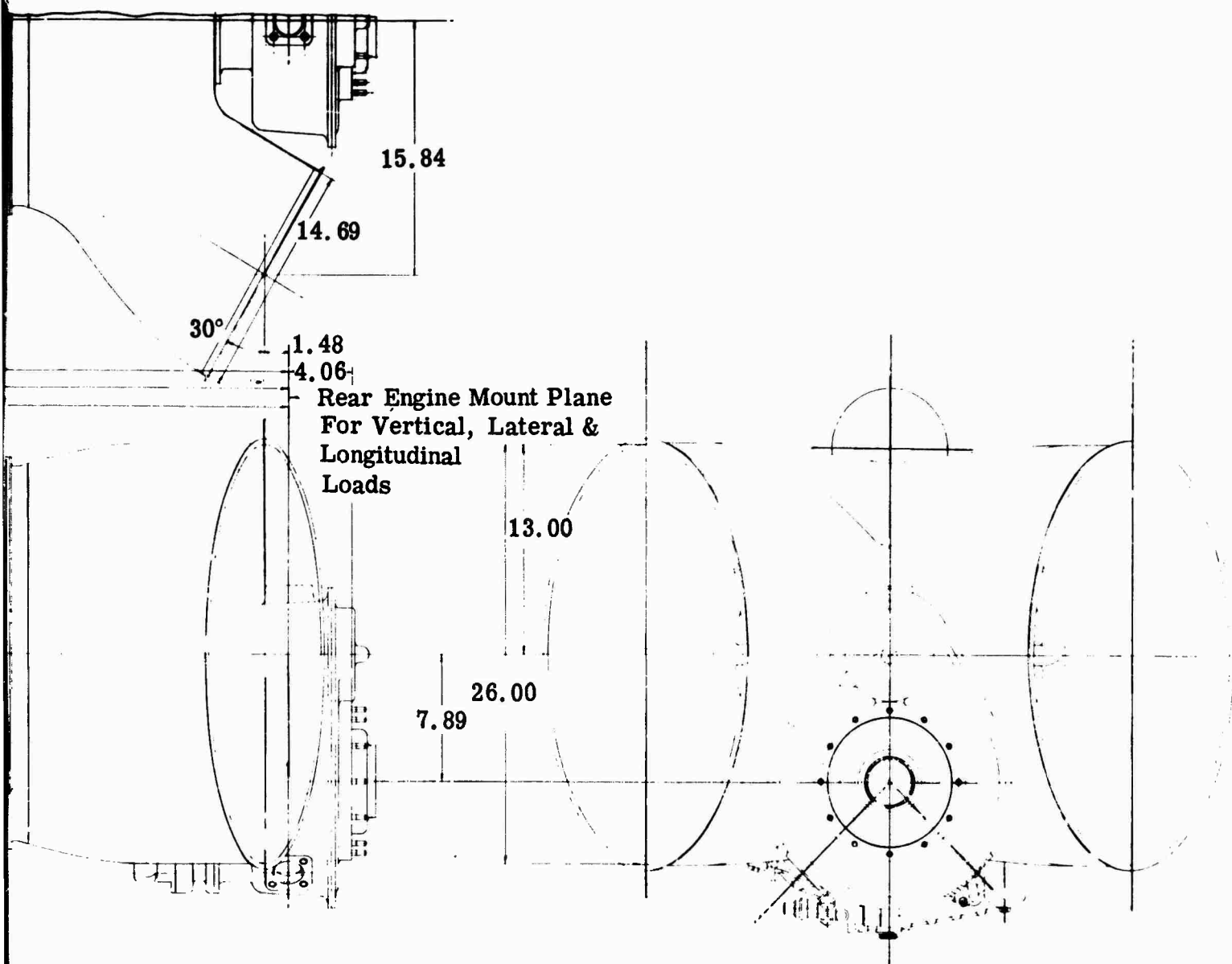
ving for Model 548-C2 (SK-10060)

2

7.600 Dia. Flange
0.24-0.25 Dia. Bolt Holes
In 7.0000 Dia. Bolt Circle
5.911-5.912 Dia. x 0.560 Deep Pilot

—4.38
Dia.

—15.75—
Dia.

—Forward Engine
Mount Pads

—Compressor Air
Inlet Housing

—Variable Geometr

— Rear E

—29.02—

—3.81

13.50

—27.00——8.16——22.54————30.70

—106.80—

—Oil Coo

—Compress

—Accessories

—Accessory Drive Housing

—Torquemeter

—Splined Output Shaft
Rotation CCW Viewed From Rear
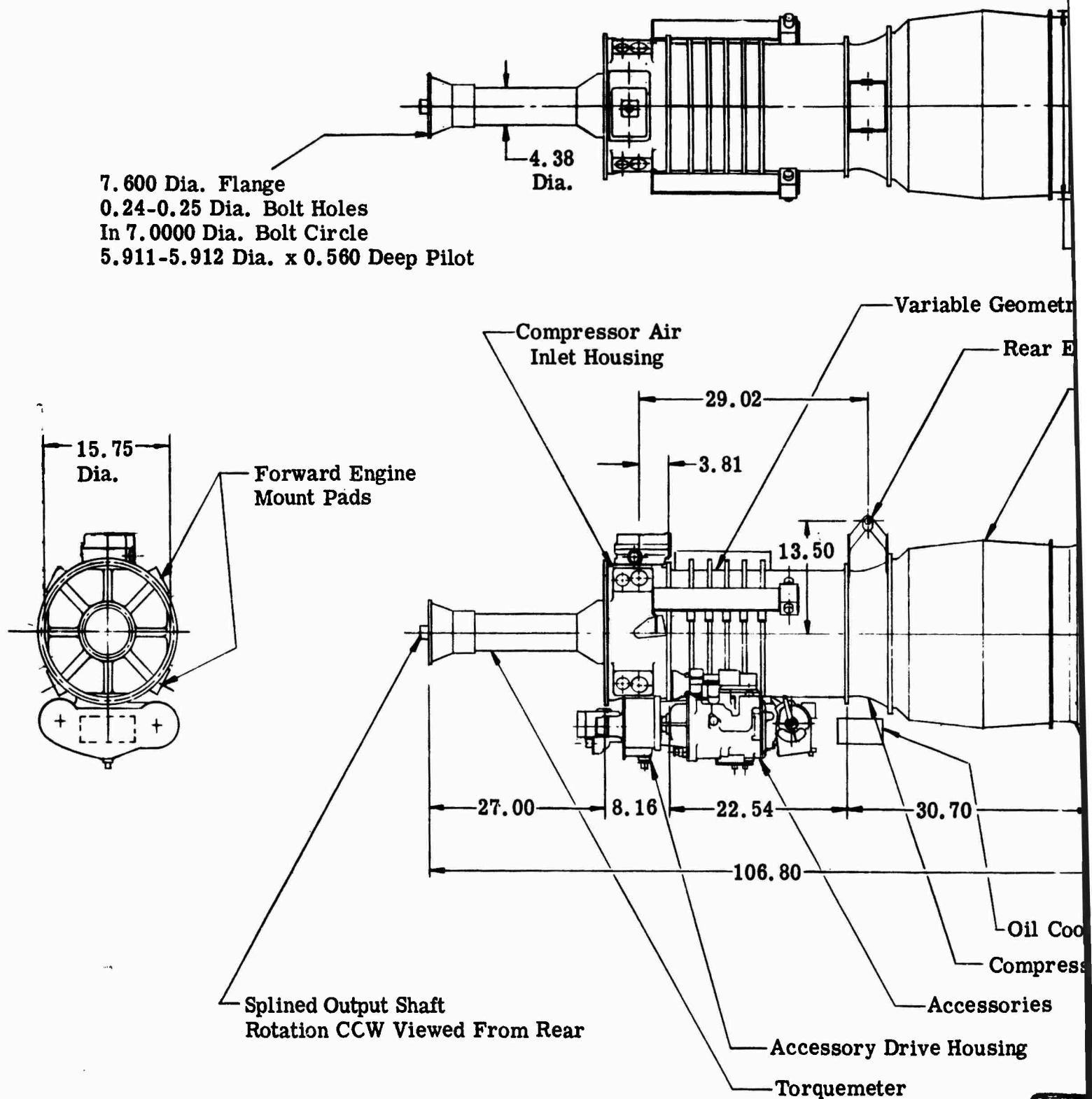
Figure 110. (U) Outline Drawing for Model 501

22.40 Dia.

Variable Geometry Compressor Assembly

Rear Engine Mount

Combustion Section Assembly

Rear Exhaust Flange

2

13.50

28.50
Dia.

28.84
Dia.

54

30.70

6.80

Oil Cooler

Turbine Unit Assembly

Compressor Diffuser Assembly
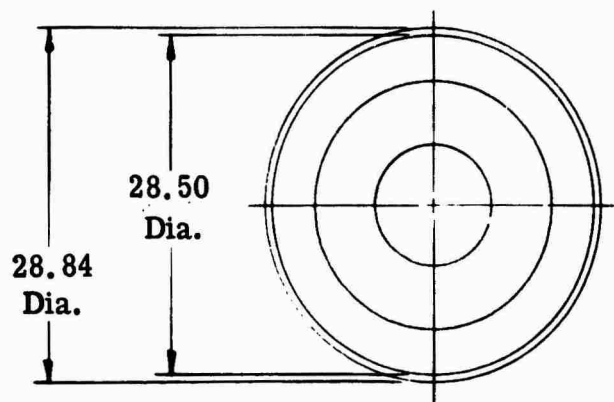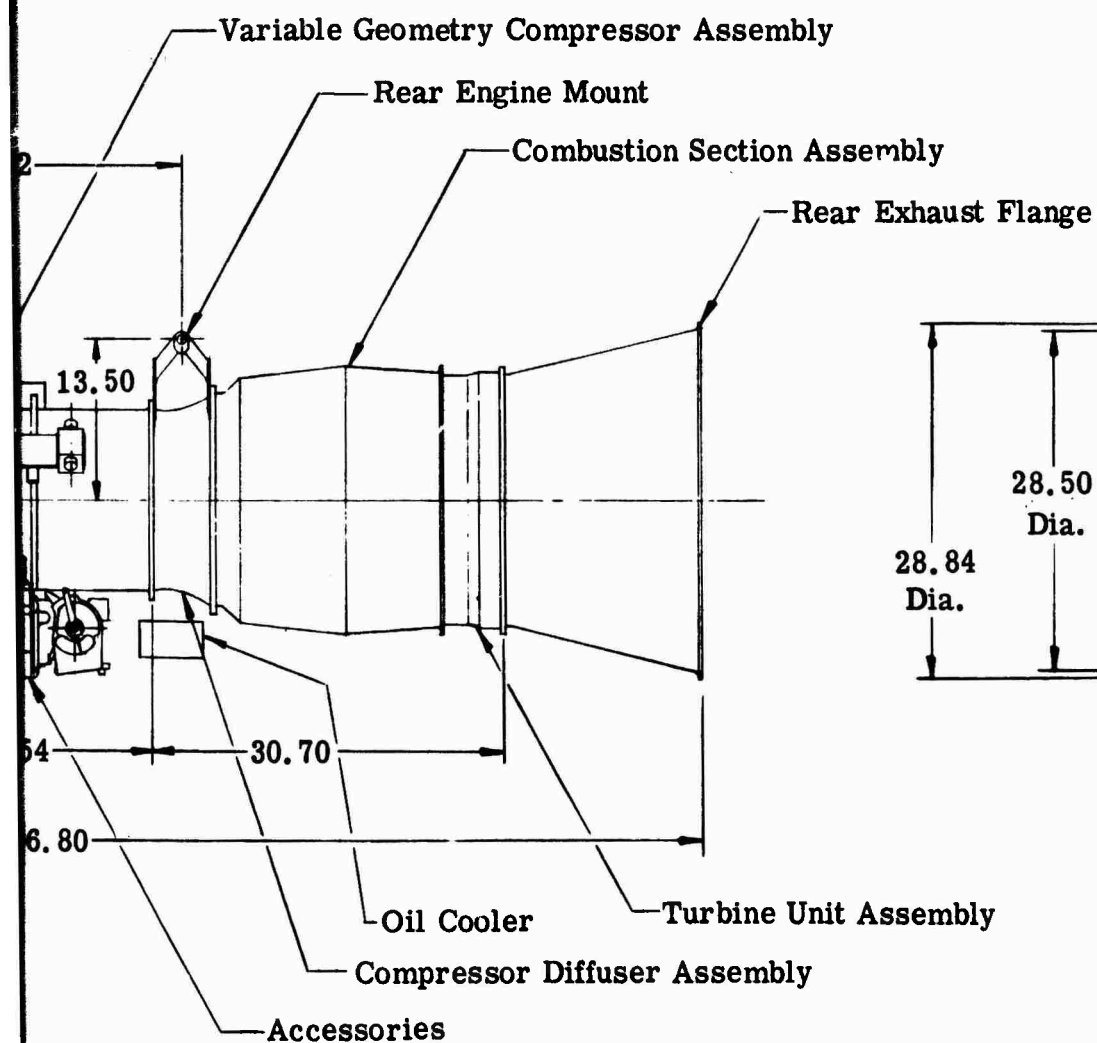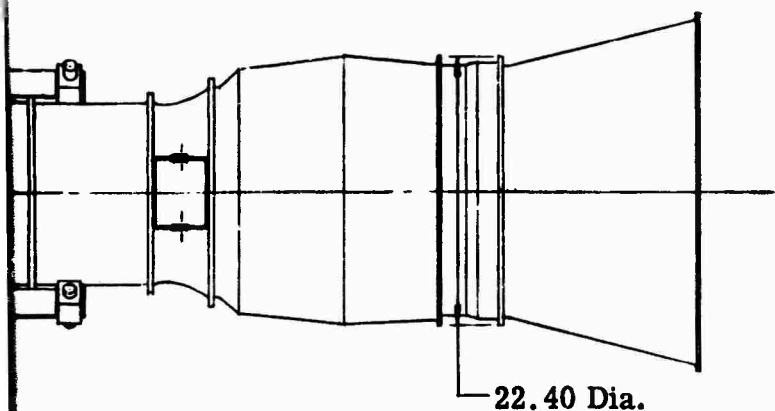
Accessories

Accessory Drive Housing

Torquemeter

ine Drawing for Model 501-M25 (SK-14614)

17.50
Dia.

Compressor Air
Inlet Housing

Variable Geometry
Compressor Assembly
Engine Mount Pads

Combustion Se

28.50

25.00
Dia.

9.00
TYP

14.
TY

3.50
8.16
22.54
23.50
Oil Cooler
72.00
77.30
Turb

Accessories

Accessory Drive Housing
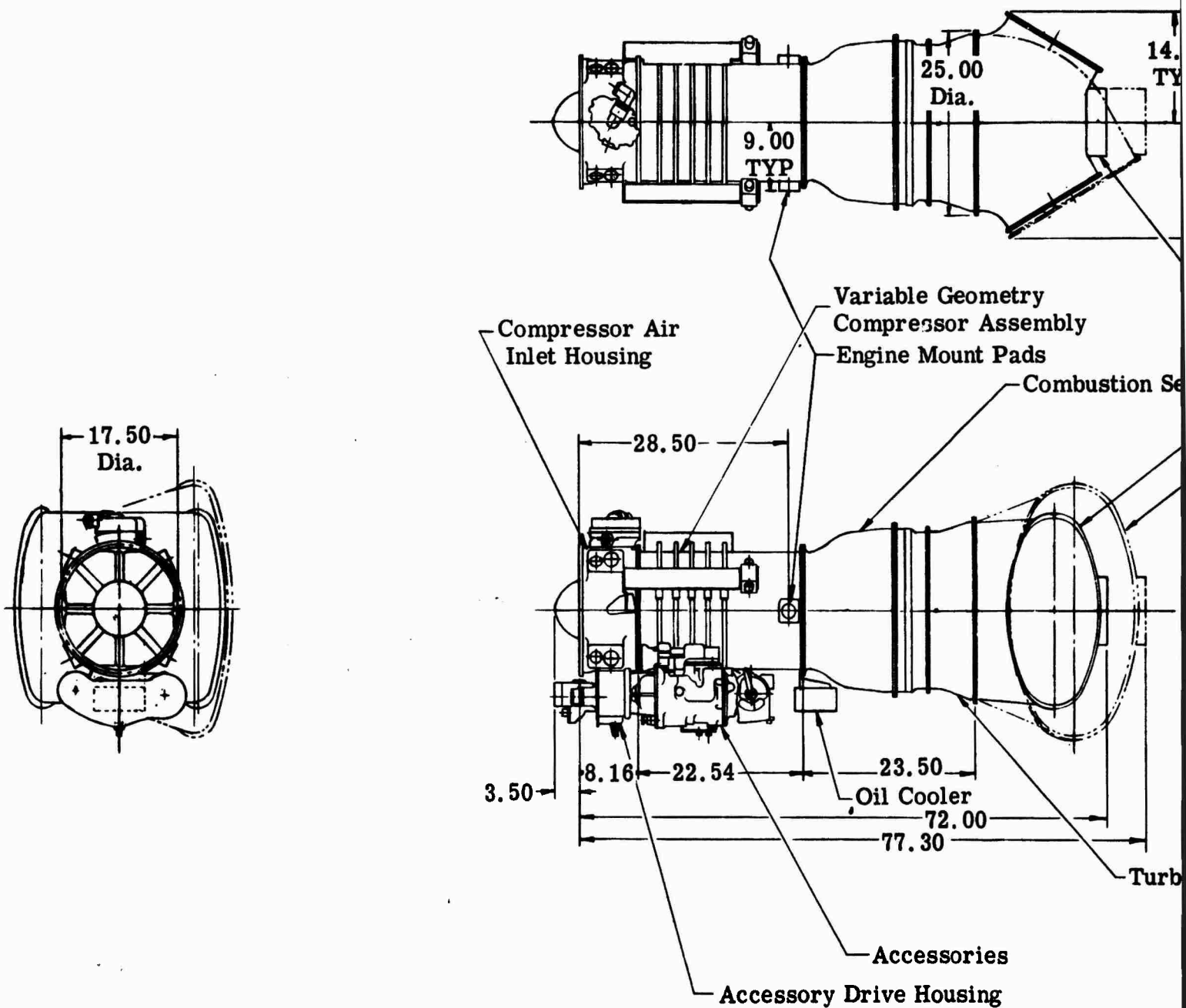
Figure 111. (U) Outline Drawing for I

199

14.50
TYP

15.00
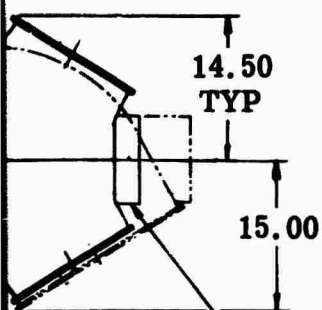
└─Rear Engine Mount

etry
sembly
ads
─Combustion Section Assembly

┌─Splined Output Drive
  Rotation CCW Viewed From Rear
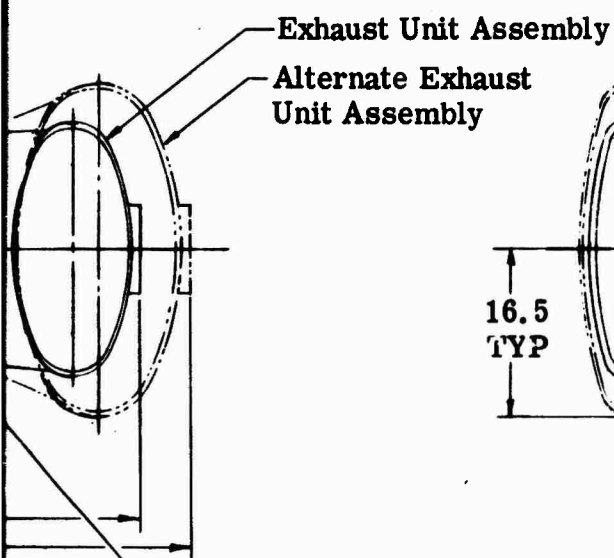
─Exhaust Unit Assembly
─Alternate Exhaust
  Unit Assembly

12.5
TYP

16.5
TYP
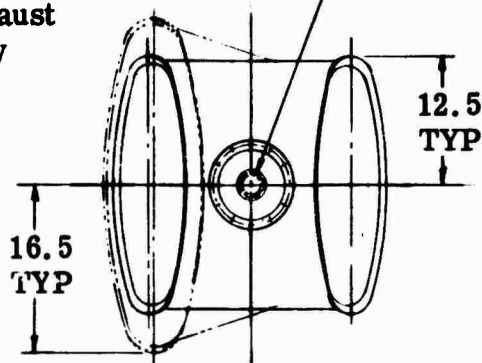
Exhaust Unit Assembly may be Rotated About
Engine ₵ to Provide Outlet Direction to Suit
Installation

└─Turbine Unit Assembly

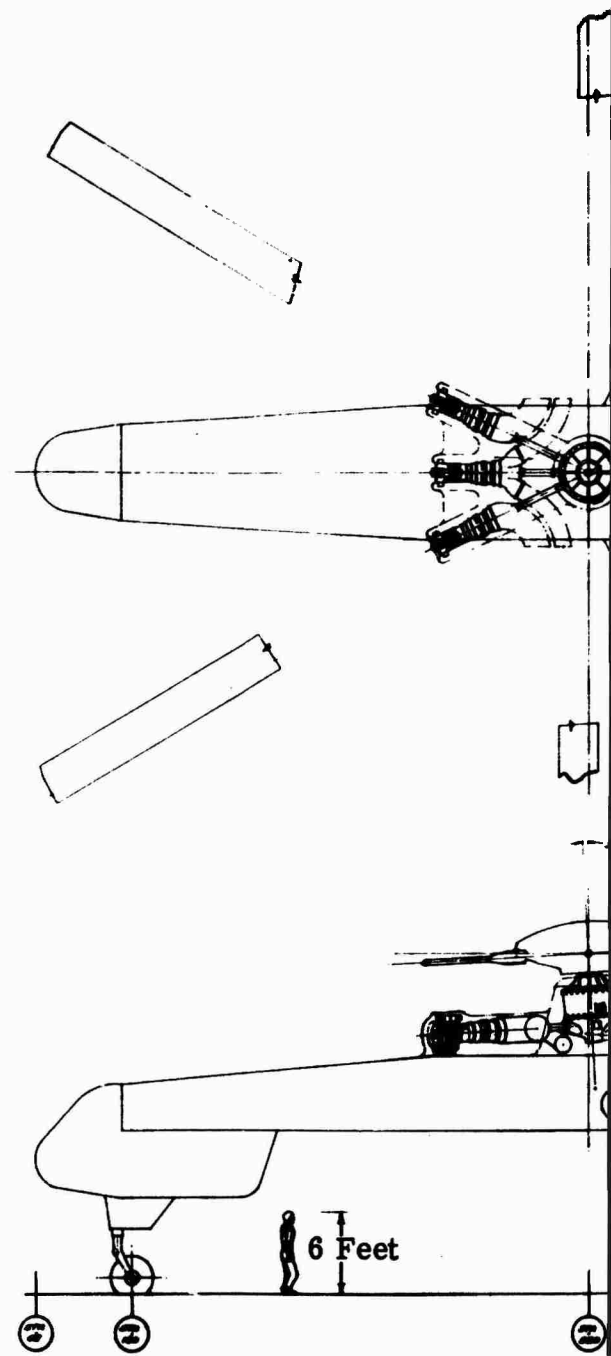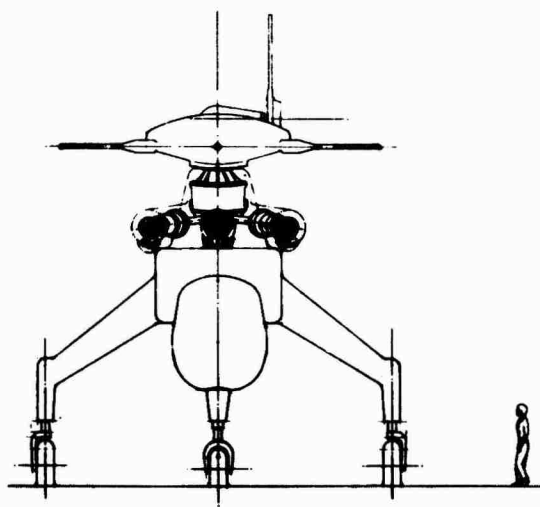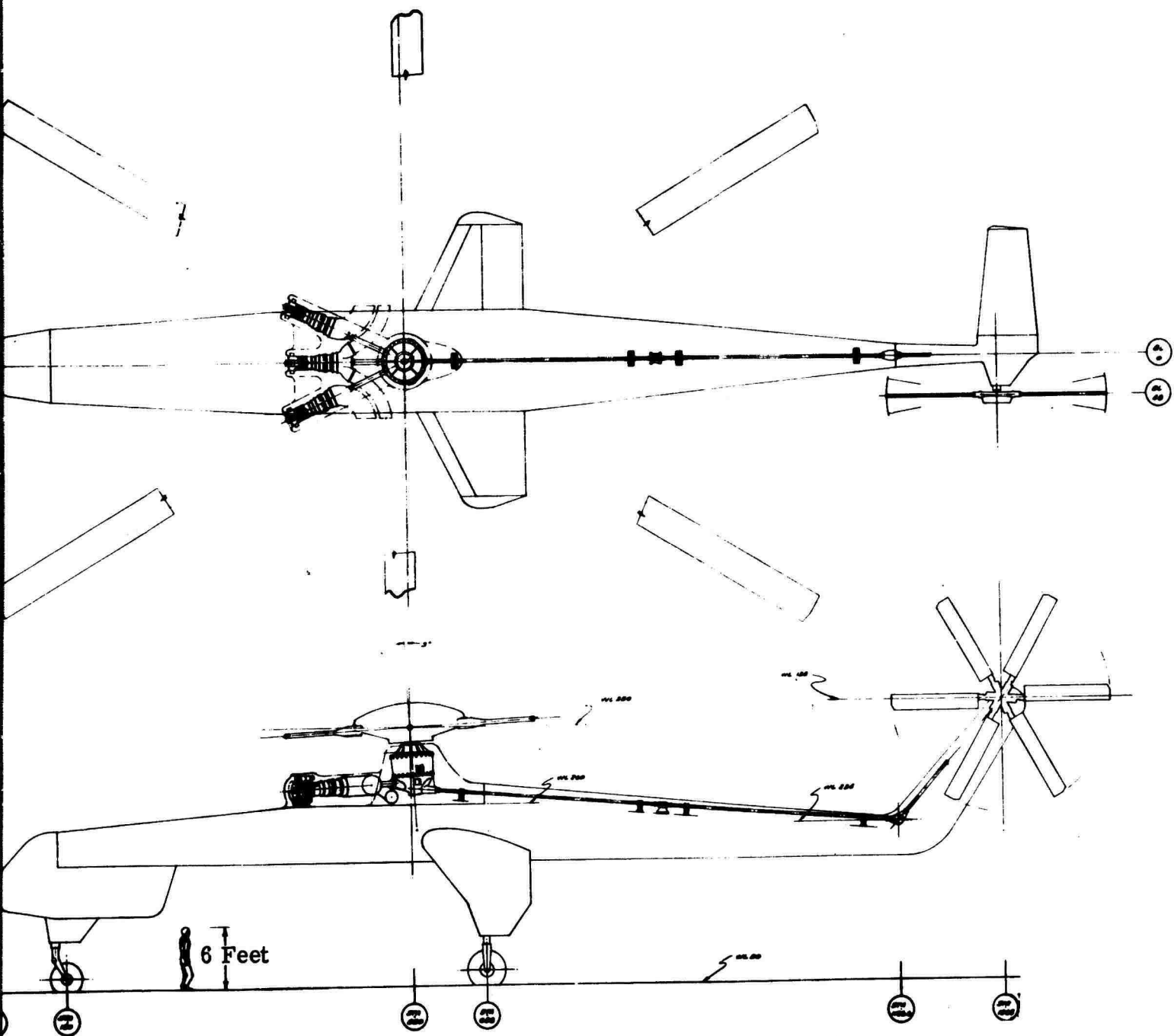ne Drawing for Model 501-M26 (SK-14873)

2

Figure 112. (U) Single-Rotor HLH, Three-Engine Mechan

gle-Rotor HLH, Three-Engine Mechanical Drive (SK-10075)

6 Feet

TABLE 14

(C) ENGINE CHARACTERISTICS AND INSTALLATION DATA (U)

Model Designation	545-C2 545-C3	546-C2 546-C3	548-C2	501-M25	501-M26
Compressor					
Number of Stages	14	14	14	14	14
Compression Ratio	11.2	11.2	11.2	11.25	9.7
Airflow, pounds per second	27.4	27.4	27.4	37.0	32.3
Turbine					
Gas Generator Stages	-	-	2	-	2
Power Turbine Stages	4	4	2	4	2
Maximum Inlet Temperature, °F.	2060	2060	2060	2060	2060
Output Speed, r.p.m.	19,320	19,320	6000*	14,300	13,000
Output Shaft Rotation**	CCW	CCW	CW	CCW	CCW
Maximum Output, Shaft Horsepower (Sea Level Static)	4100	4500	4480	6000	5450
Engine Weight, pounds	1141	700	822***	955	1030
Oil Flow, pounds per minute					
Power Section	29	29	29	34	34
Reduction Gear	-	-	60	-	-
Heat Rejection, B.t.u. per minute					
Power Section	700	700	700	1400	-
Reduction Gear	-	-	1200	-	-
Compressor Inlet Duct Inside Diameter, inches	15.5	15.5	15.5	15.75	15.75

*Gas producer speed of 19,320 r.p.m.
**When viewed from rear
***732 pounds for 19,320 r.p.m. direct drive

Note:
CCW counterclockwise
CW clockwise

(U) HLH Tandem-Rotor, Three-Engine Mechanical Drive

(U) Figure 113 shows a tandem-rotor helicopter in which the engine arrangement differs slightly from that of the single-rotor installation previously discussed. The lower half of the rear-rotor transmission is identical to that used in the single-rotor configuration. A 180-degree turning box for the synchronizing shaft is mounted on the takeoff pad used for the antitorque rotor drive shaft of the single-rotor aircraft.

(U) The upper half of the transmissions, containing the planetary systems, is the same for both the forward and aft rotors. Size and weight of these upper case assemblies were reduced proportionately when compared with the corresponding single-rotor configuration. Each assembly must be designed for only about 60 percent of the total engine power.

(U) The synchronizing shaft is fitted externally to the aircraft structure and is covered with a lightweight shroud. External placement is provided to permit ease of access for servicing and shafting changes.

(U) Compact clustering of the engines around the main transmission is also a feature of this configuration.

(U) HLH Tandem-Rotor, Four-Regenerative-Engine Mechanical Drive

(U) Figure 114 shows a tandem-rotor helicopter with four front-drive Model 545 regenerative turboshaft engines. The arrangement provides good inlet air conditions to the engines and minimizes exhaust gas ingestion. The forward transmission and the upper half (planetary system) are of the same design shown in Figure 113. The lower half of the main transmission is, of course, different than that shown in Figure 113. Placement of the synchronizing shafting is the same as that for the three-engine configuration.

(U) Engine cowling and the inlet duct are arranged in such a manner that servicing and engine changes can be readily made. Engine exhaust is directed overboard without the use of ducting. The total engine weight for this configuration is 4460 pounds. The Models 545, 546, and 501-M25 turboshaft engines will fit within the envelopes shown in this arrangement.
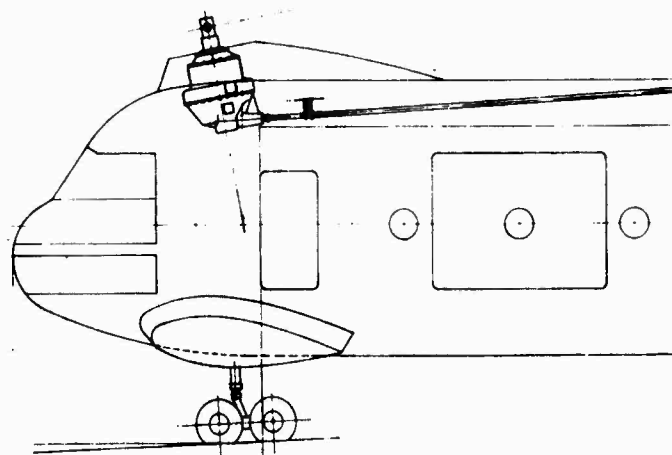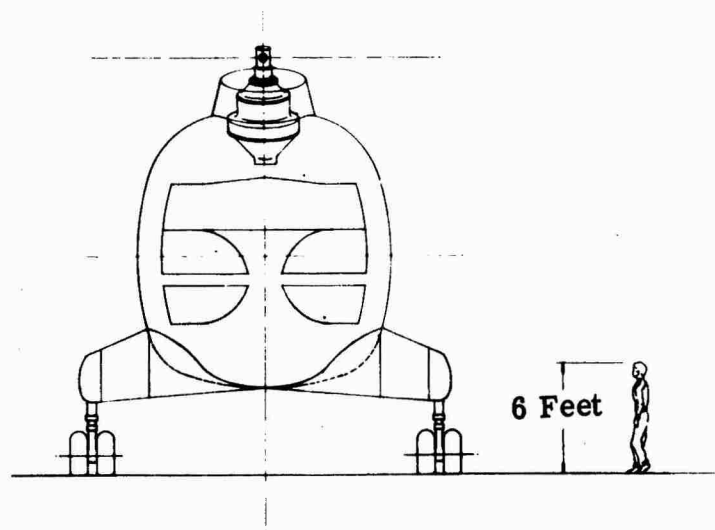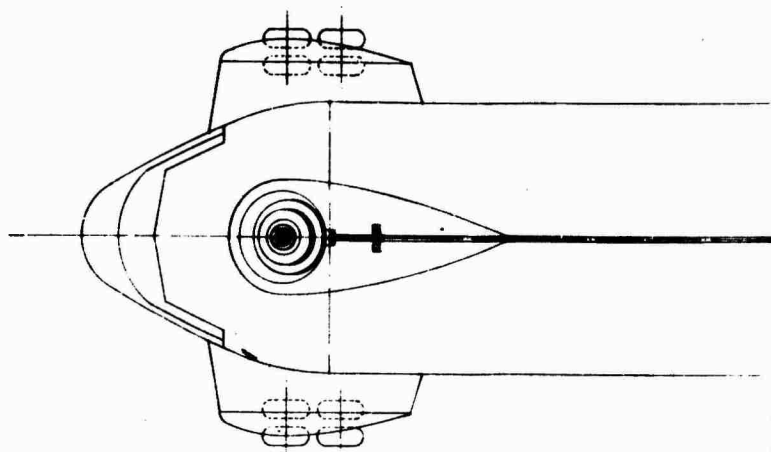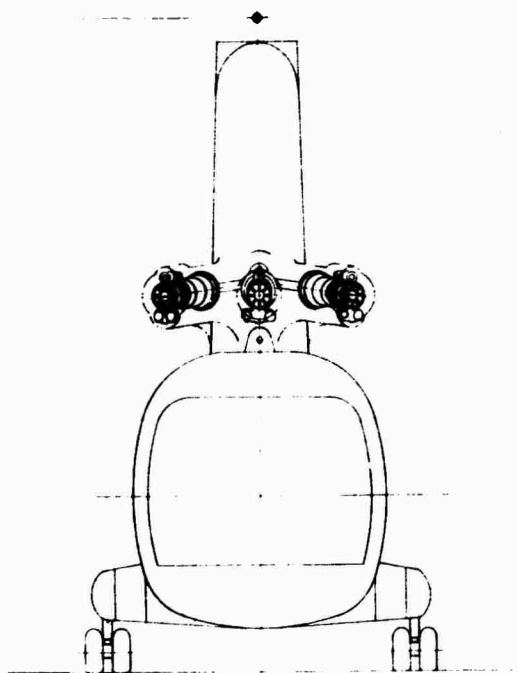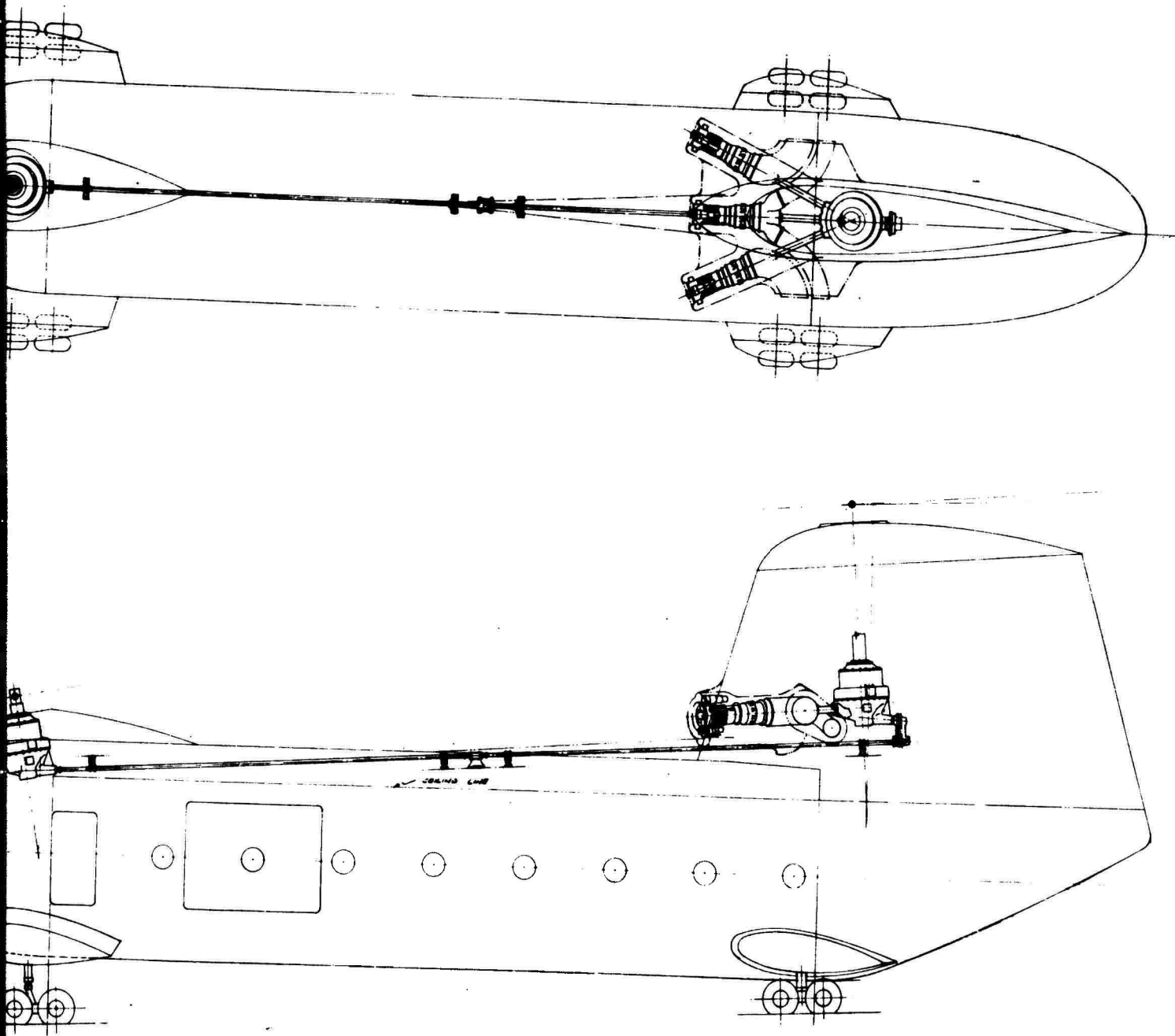
Figure 113. (U) Tandem-Rotor HLH, Three-Engine Med

-Rotor HLH, Three-Engine Mechanical Drive (SK-10076)
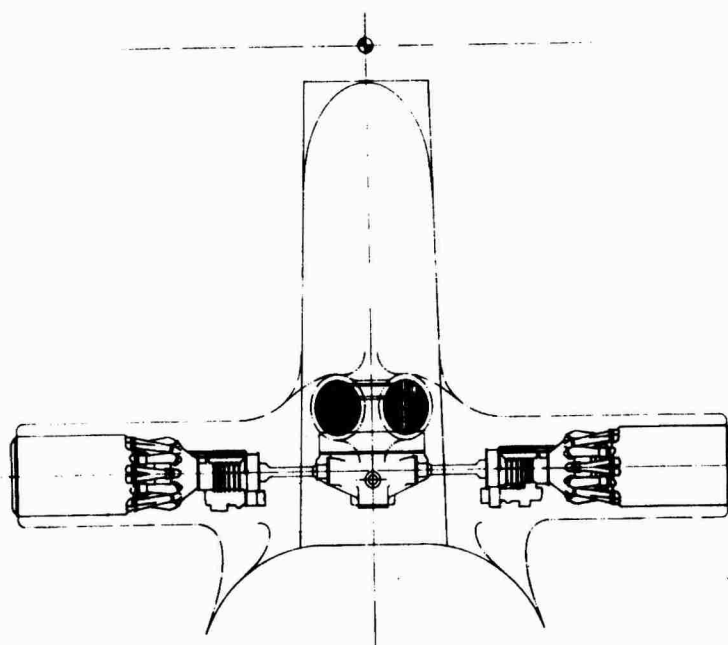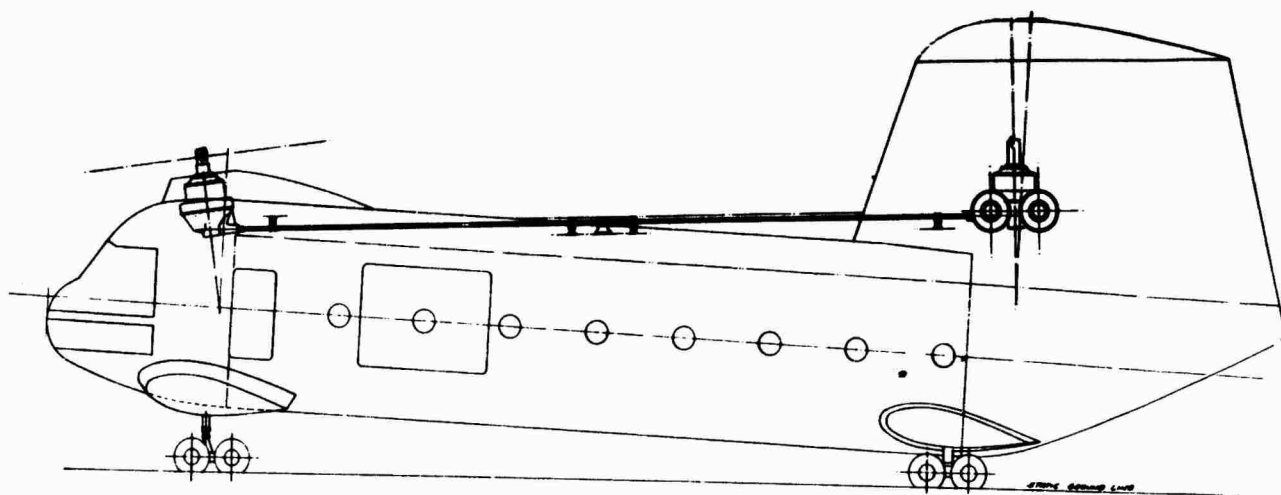
Figure 114. (U) Tandem-Rotor HLH, Four-Regenerative Engine Me

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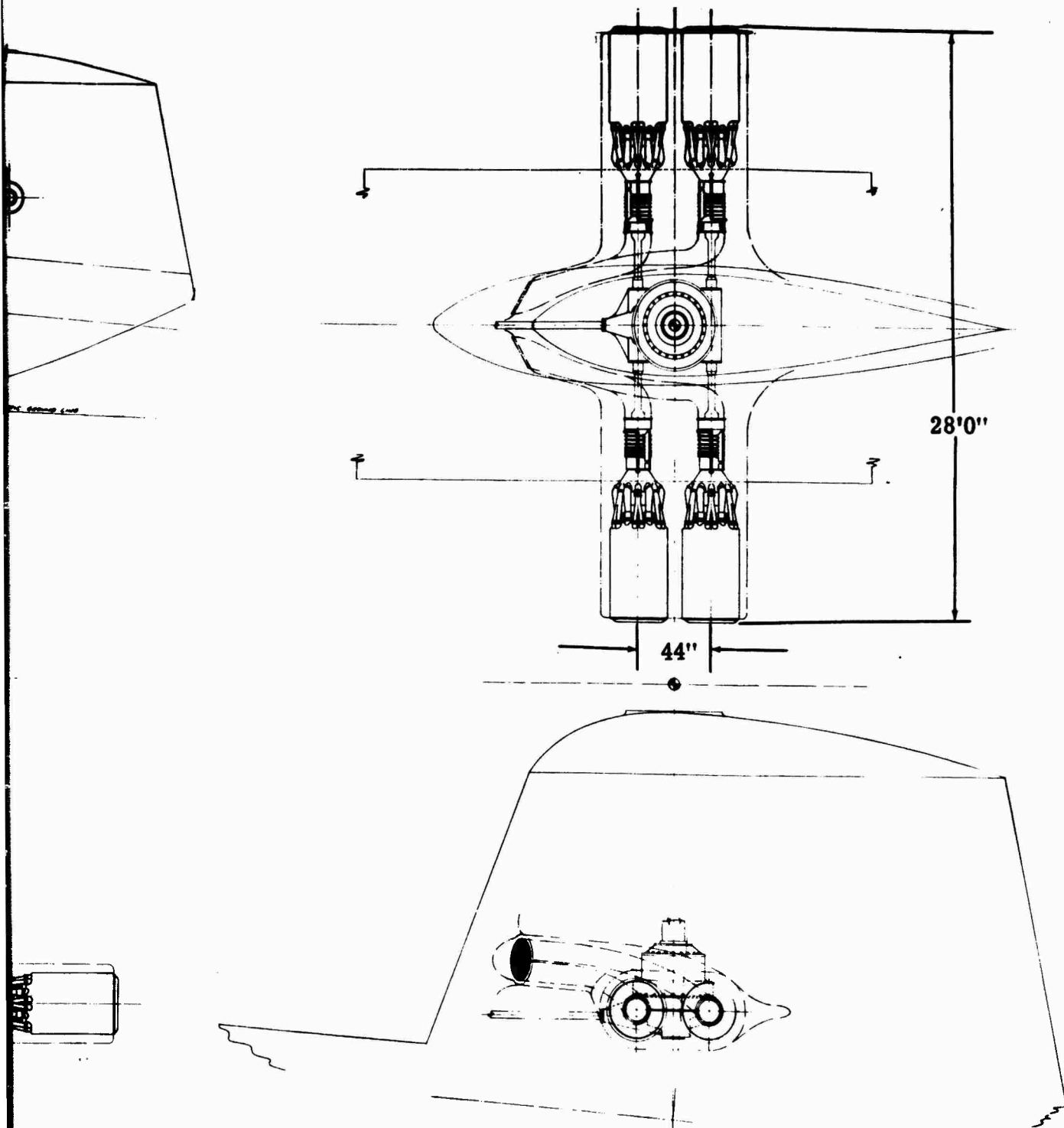
Four-Regenerative Engine Mechanical Drive (SK-10077)

(U) Integrated Power-Turbine Transmission Drive

(U) Figure 115 shows an integrated power-turbine transmission drive system as installed in a single-rotor version of the HLH. This configuration, with minor modification of the takeoff pad for the synchronizing shaft, can be used in the tandem-rotor airframe type. Details of the transmission design are presented in the section titled Power Coupling Studies.

(U) Gas producers (546 or 501 series) are very compactly arranged around the integrated power-turbine transmission system. The gas producers are readily accessible for maintenance and engine changes. Total powerplant weight, including three gas producers, ducting, three power turbines, main transmission, and supporting structure as installed in the single-rotor helicopter, is estimated at 5700 pounds.

Figure 115. (U) Single-Rotor HLH, Integrated Engine-Power Turbine

grated Engine-Power Turbine Transmission Drive System (SK-10082)

6 Feet

(U) DEVELOPMENT REQUIREMENTS

Variations of two basic engines (Models T56 and T78) which are now in production or under development were considered during this study. The Mode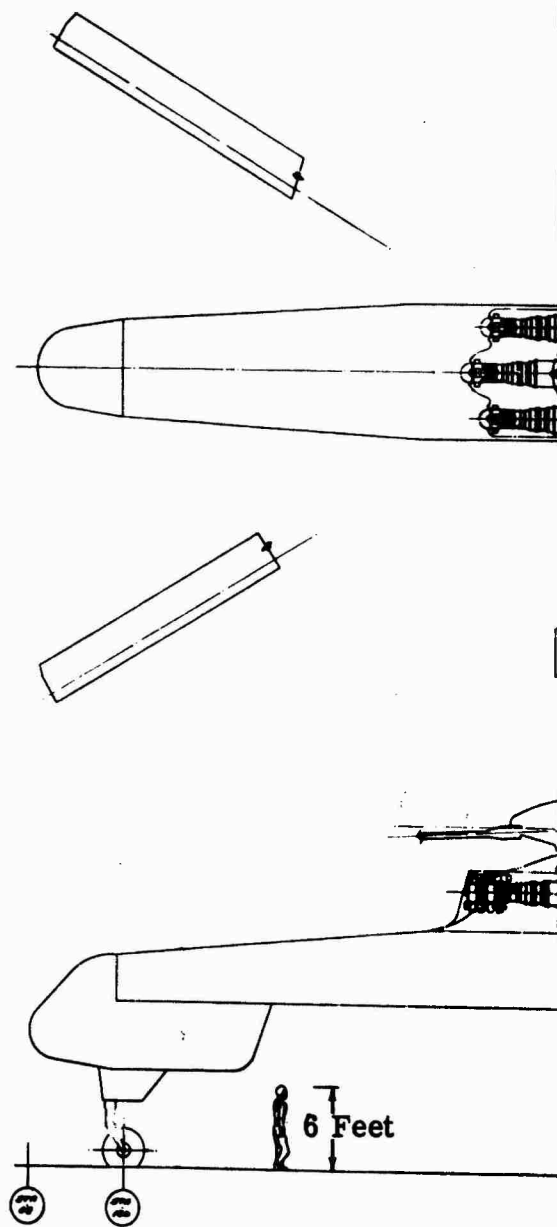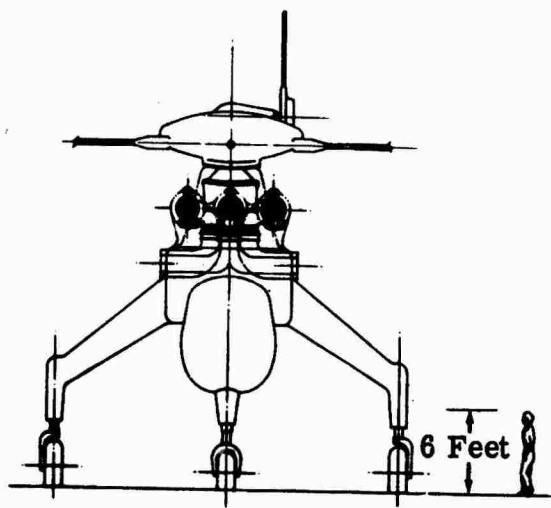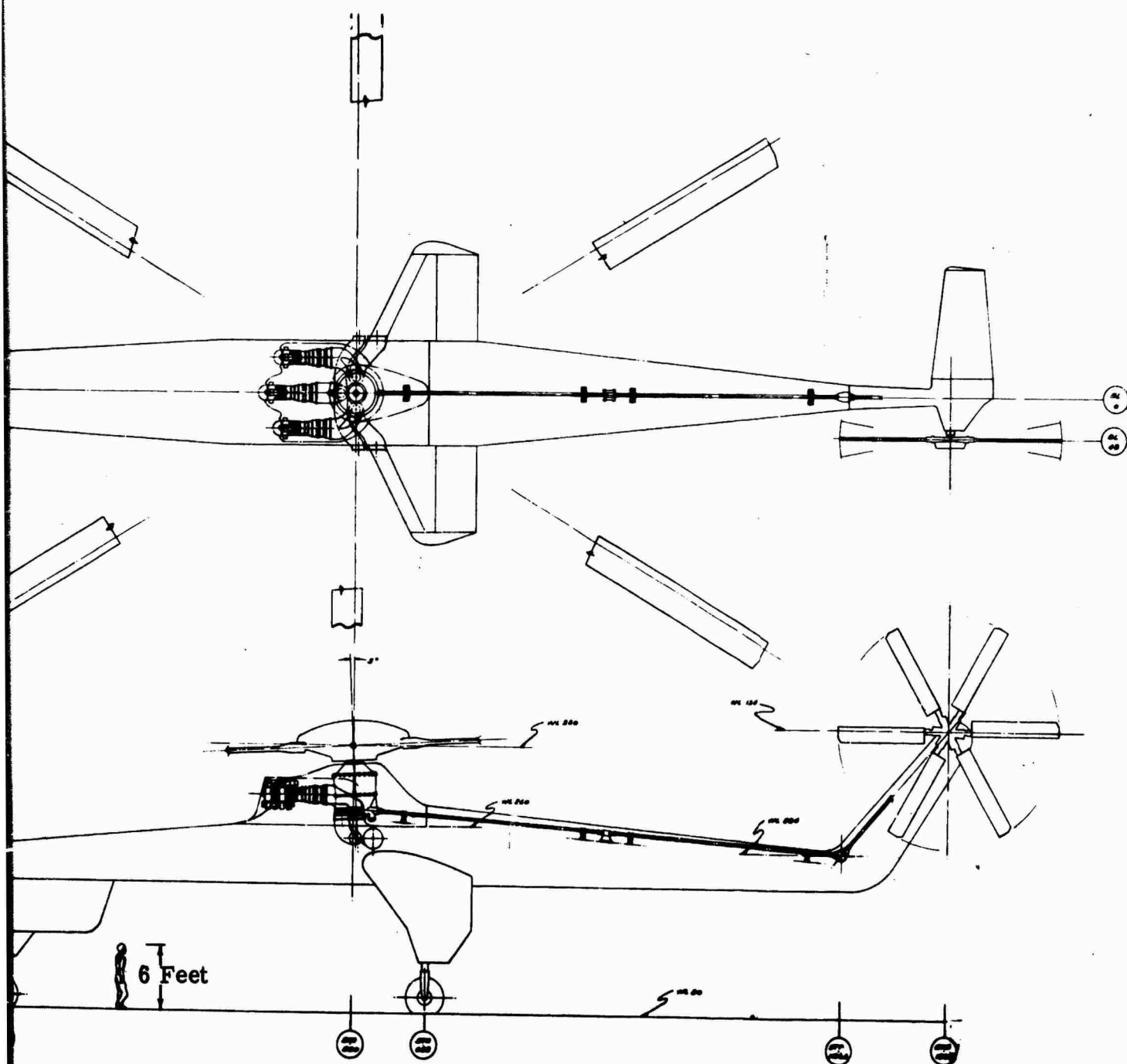l T56, which has been in production since 1954, now powers the C130, P3A, E2A, Electra, and Convair commercial aircraft. The Model T78 is now under development on Navy contract and is scheduled to complete its preliminary flight rating test (PFRT) by December 1966. Model qualification testing (MQT) of the Model T78 is scheduled for completion by April 1968. The development requirements of the various engines considered in this study are discussed in the following paragraphs.

MODEL 501-M25 (FIXED-TURBINE T56)

This engine is similar to the T56-A-15 which is now in production. The T56-A-15 has a maximum turbine inlet temperature rating of 1970°F. The 90°F. increase in turbine inlet temperature specified for the M25 should not impose a significant development problem. Another deviation from the production configuration is the use of variable compressor geometry. A variable geometry compressor of the type to be used in the M25 engine has been built and tested; however, some additional development will be required in this area. In addition, the control system will require modification.

It is anticipated that a PFRT program could be completed within 24 months from go-ahead. The MQT could be completed within 30 months from go-ahead.

MODEL 501-M26 (FREE-TURBINE T56)

This engine is similar to the Model 501-M25, and, as such, similar development problems are anticipated. In addition, this model includes a free turbine. Design and development of this component are considered to be straightforward engineering problems. Although additional engineering and development costs would be incurred, ··· significant effect on availability is anticipated.

MODELS 545-C2 AND 545-C3 (FIXED-TURBINE T78)

These engines are almost identical to the baseline Model T78 (Allison Model 545-B2) except that the exhaust nozzle is larger on the C2 and C3 to optimize the engine for a helicopter. No new development problems, therefore, are anticipated.

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The Model 545-C3 engine operates at constant turbine inlet temperature and constant engine speed throughout the power range. The power is modulated on the 545-C3 by varying the compressor variable geometry; however, this is not expected to cause any major developmental problems.

The minor variations from the basic Model T78 engine are not expected to add over 3 months to the basic T78 schedule.

MODEL 546-C2 (FIXED TURBINE-NONREGENERATIVE T78)

The 546-C2 engine is a nonregenerative version of the 545-C3 engine. Therefore, the 546-C2 engine has some additional features which are not common with those of the Model T78 engine. The nonregenerative engine requires a new diffuser, since the diffuser on this engine ducts the air directly from the compressor to the combustor. The turbine is reset to provide the additional expansion ratio required of the nonregenerative turbine. These changes are not expected to cause any major problems because the major portion of the engine is identical to the basic Model T78 engine. Design and development of the new turbine to PFRT status would require approximately 24 months. Development to fully qualified status would require an additional 6 months.

MODEL 548-C2 (FREE-TURBINE NONREGENERATIVE T78)

The Model 548-C2 engine is a nonregenerative, free turbine version of the Model T78 engine. This engine will incorporate a new diffuser and a reset in the turbine aerodynamic configuration plus the separation of the first and last two stages of the turbine. With the two separate turbines, the engine will require new bearings for the free turbine with the related bearing supports. Development time requirements would be very similar to those of the Model 546-C2.

GAS-COUPLED ROTOR DRIVE

The gas-coupled rotor drive modification of the T78 would be similar in scope to that for a free turbine configuration except the power turbine would be integrated with the transmission. Lubrication and power turbine rotor balance system changes would be required. Development time and availability would be very similar to that for the Model 548-C2 engine.

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APPENDIX

(C) MISSION ANALYSIS METHOD (U)

(U) This appendix describes in detail the procedure used in the mission studies. (See section titled Performance Studies.) Four Model 548-C2 engines were installed in the vehicle used for the method illustration. The heavy-lift, transport, and ferry missions are considered.

(U) To evaluate engine performance in a helicopter application, the following curves are necessary:

- Shaft horsepower versus fuel flow with lines of velocity for the Model 548-C2 engine (Figure 116)
- Total shaft horsepower versus velocity with lines of design gross weight including a range of frontal areas (See Figure 1)
- Specific range (velocity/fuel flow) versus velocity with lines of gross weight for four-, three-, and two-engine operation (Figures 117 through 119)
- Specific range versus gross weight with lines of velocity (Figure 120)
- Range versus gross weight (Figure 121) with lines of velocity obtained by integrating the areas under the curves presented in Figure 120
- Specific range versus gross weight at maximum range for four-, three-, and two-engine operation (Figure 122)
- Shaft horsepower versus gross weight for various velocities and 100-feet-per-minute rate of climb (Figure 123)

(U) Figure 123 was used to indicate when to shut off an engine during each mission. The helicopter was required to have a 100-feet-per-minute rate of climb at all times. As shown in Figure 123, the normal power rating for one engine intersects the 100-feet-per-minute rate of climb curve at a gross weight of approximately 52,000 pounds. This was used as a basis for discontinuing one of the three engines during the ferry mission. Also, on the inbound leg of the heavy-lift and transport missions, with the payload removed, only two engines were used. It was not possible to shut down to one engine because of safety considerations.

(U) A detailed summary of the results of the mission analysis using the Model 548-C2, is presented in Table 15. The procedure used for the analysis of each mission is presented in the following paragraphs.

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(C) HEAVY LIFT MISSION (U)

(U) As stated in the Mission Analysis subsection, an empty weight/ design gross weight ratio fo 0.47 for the heavy-lift mission was considered to be a reasonable estimate. Therefore, the following expressions were used to arrive at a solution:

$$EW/DGW = 0.47$$
$$DGW = EW + W_p + W_f + W_c$$

where

DGW	= Design Gross Weight
EW	= Empty Weight
W_p	= Payload (40,000 pounds)
W_f	= Fuel Weight
W_c	= Crew and Trapped Fluid Weight (700 pounds)

(U) An iterative process was used which consisted of the following steps.

1. Assume a design gross weight.

2. Calculate fuel requirements for the outbound phase of the mission.

3. Reduce the assumed design gross weight by subtracting the fuel used in the outbound phase of the mission.

4. Remove the payload.

5. Calculate fuel requirements for the inbound phase of the mission.

6. Subtract fuel used on inbound phase of mission.

7. Subtract crew and trapped fluid weight to arrive at empty weight.

8. Iterate until empty weight/design gross weight equal 0.47.

(C) The final iteration showed that the design gross weight of an HLH using four Model 548-C2 engines would be 83,700 pounds. The values

TABLE 15
(C) MODEL 548-C2 MISSION STUDY DATA (U)

	Tons Payload	HLH Weight (lb.)*	Total Shaft Horsepower	Shaft Horsepower per Engine	Time (hr.)
Heavy Lift Mission (20-Nautical Mile Radius)					
Start, Warm-up, Takeoff	20	83,700	—	—	0.033
Hover at Takeoff	20	83,500	12,500	3,125	0.083
Cruise (Outbound)	20	82,949	7,720	1,930	0.211
Hover at Destination	20	81,959	12,200	3,050	0.167
Cruise (Inbound)	0	40,879	6,920	3,460	0.154
Reserve (10 percent) Crew (700 lb.)	—	40,333	—	—	—
Empty Weight	—	39,296	—	—	—
Transport Mission (100-Nautical Mile Radius)					
Start, Warm-up, Takeoff	12	71,960	—	—	0.033
Hover at Takeoff	12	71,760	10,320	2,582	0.050
Cruise (Outbound)	12	71,472	6,510	1,627	0.909
Hover at Midpoint	12	67,632	9,520	2,380	0.033
Cruise (Inbound)	0	43,451	6,930	3,465	0.769
Reserve (10 percent) Crew (700 lb.)	—	40,720	—	—	—
Empty Weight	—	39,296	—	—	—
Ferry Mission (1500-Nautical-Mile Range)					
Start, Warm-up, Takeoff	0 (Fuel only)	103,716	—	—	0.033
Cruise	—	100,258	—	2,690	1.11
	—	74,500**	—	2,645	10.32
	—	50,658**	—	2,515	0.95
Reserve (10 percent)	—	—	—	—	—

*HLH weight at initiation of mission phase, unless otherwise specified.
**Average HLH weight for each phase.

TABLE 15
(C) MODEL 548-C2 MISSION STUDY DATA (U)

HLH Weight (lb.)*	Total Shaft Horsepower	Shaft Horsepower per Engine	Time (hr.)	Velocity (knots)	Fuel Flow per Hour per Engine	Fuel Weight (lb.)	Engines Operating
3,700	—	—	0.033	0	—	200	4
3,500	12,500	3,125	0.083	0	1,652	551	4
2,949	7,720	1,930	0.211	95	1,175	990	4
1,959	12,200	3,050	0.167	0	1,620	1,080	4
0,879	6,920	3,460	0.154	130	1,775	546	2
0,333	—	—	—	—	—	337	-
9,296	—	—	—	—	—	—	-
					Total	3,704 lb.	
1,960	—	—	0.033	0	—	200	4
1,760	10,320	2,582	0.050	0	1,440	288	4
1,472	6,510	1,627	0.909	110	1,056	3,840	4
7,632	9,520	2,380	0.033	0	1,360	181	4
3,451	6,930	3,465	0.769	130	1,775	2,731	2
0,720	—	—	—	—	—	724	-
9,296	—	—	—	—	—	—	-
					Total	7,964 lb.	
3,716	—	—	0.033	0	—	200	-
0,258	—	2,690	1.11	125	1,472	6,516	4
4,500**	—	2,645	10.32	122	1,450	45,000	3
0,658**	—	2,515	0.95	105	1,404	2,684	2
—	—	—	—	—	—	5,430	—
					Total	59,830 lb.	

nless otherwise specified.

AL

2

obtained in this iteration are as follows. With a gross weight of 83,700 pounds and a 200-pound fuel allowance for the 2-minute start, warm-up and takeoff phases, the gross weight at hover during takeoff is 83,500 pounds. Using Figure 124 for the hover power required, a total shaft horsepower of 12,500 is obtained. Since four engines are in use, the shaft horsepower per engine is 3125. From Figure 116, a fuel flow per engine of 1652 pounds per hour is obtained. Thus, for 5 minutes of hover time, the fuel used is 551 pounds. The initial outbound cruise gross weight is only 82,949 pounds. As shown in Figure 121, with a velocity of 95 knots and a 200-square-foot frontal area for a 20-nautical-mile mission, the fuel used will be 990 pounds. The hover at destination requires 10 minutes at a gross weight of 81,959 pounds. Using Figure 124, the total shaft horsepower required is 12,200 or 3,050 shaft horsepower per engine. Figure 116 gives the fuel flow per engine as 1,620 pounds per hour. Therefore, for hover at destination the total fuel used is 1,080 pounds. At this point, the payload is removed so that the final cruise inbound gross weight is 40,879 pounds. Since shutdown of an engine may occur at some gross weight under 52,000 pounds, only two engines are used on this portion. The fuel used was obtained directly from Figure 121, using the 130-knot velocity line. In this case, the fuel used is 546 pounds. The total fuel used on the mission is 3,367 pounds. A 10-percent reserve of 337 pounds is added to this to get the total fuel required—3704 pounds. An empty weight of 39,296 pounds, as shown in Table 15, results when the fuel, payload, and crew and trapped fluid weights are subtracted from the design gross weight. The resulting empty weight/design gross weight is 0.47 and the iteration is complete. This empty weight of 39,296 pounds was used for the transport and ferry missions.

(C) TRANSPORT MISSION (U)

(C) The procedure for establishing transport mission initial gross weight and fuel requirements was similar to that used for the heavy-lift mission. However, instead of closing the iteration on empty weight/design gross weight it was closed on helicopter empty weight. The final iteration showed the transport mission initial gross weight to be 71,960 pounds and to require 7,964 pounds of fuel.

(C) The values obtained in the final iteration are as follows. The start, warm-up, and takeoff fuel allowance was 200 pounds. Therefore, hover gross weight was reduced to 71,760 pounds. Using Figure 124, it was found that a total shaft horsepower of 10,320 (2,582 shaft horsepower per engine) was required. Figure 116 shows that a fuel flow of 1,440

pounds per hour per engine is required. Thus, for a 3-minute hover period with four engines operating, 288 pounds of fuel are consumed. Figure 121 shows that 3,840 pounds of fuel are required for the 110-knot outbound cruise, with an initial gross weight of 71,472 pounds. Using Figure 124, the horsepower required to hover at midpoint with a gross weight of 67,632 pounds was found to be 9,520 total or 2,380 horsepower per engine. Figure 116 shows that a fuel flow of 1,360 pounds per hour per engine is required for a total fuel consumption of 181 pounds during the hover. The 12-ton payload was removed and the inbound cruise was initiated with a gross weight of 43,451 pounds. As in the heavy-lift mission, it was possible to shut down engines on the inbound cruise. Thus, for two engines and 130-knots cruise speed, 2,731 pounds of fuel are consumed, as shown on Figure 121. The reserve fuel weight was obtained by increasing by 10 percent the 7,240 pounds used for a total fuel load of 7,964 pounds at mission initiation. When the payload, fuel, and crew and trapped fluid weights are subtracted from the gross weight, an empty weight of 39,296 pounds results, thus completing the iteration.

(C) FERRY MISSION (U)

(C) The 1500-nautical-mile mission is divided into three parts. The first portion is flown with four engines operating. The second portion is flown with three engines operating and one engine shut down. (Three engines are used when the weight is between 97,000 pounds and 52,000 pounds.) The third portion is flown with two engines operating and two engines shut down. The average gross weight for each potion of the mission was used to determine the fuel used and the distance traveled. This is possible since the velocity/fuel flow ratio maximum range curves are linear as shown in Figure 122.

(C) The method of calculation (Table 16) requires an initial estimate of fuel for the entire mission. The value used was 53,900 pounds. Step 5—which is start, warm-up, and takeoff—requires 200 pounds of fuel. For reserve fuel, 10 percent of the fuel required was used. Additional tankage was provided for the ferry mission and its weight was established as 7.5 percent of the ferry mission fuel less transport mission fuel weight. An aircraft gross weight of 103,371 pounds was used. Since 200 pounds of fuel was used before becoming airborne, the gross weight at flight initiation was 103,171 pounds and the average gross weight for the initial phase of the four-engine operation was 100,085 pounds. With this average gross

weight, a velocity/fuel flow ratio of 0.0212 will be obtained. (See Figure 122.) By multiplying the delta gross weight by this value, a distance of 131 miles is obtained.

TABLE 16
(C) METHOD OF FERRY-MISSION CALCULATION WITH
TANKAGE* INCLUDED (U)

Step		Iteration	
		Initial	Final
1	Empty Weight, lb.	39,296	39,296
2	Crew, lb.	700	700
3	Subtotal, lb.	39,996	39,996
4	Estimated Fuel, lb.	53,900	54,200
5	Starting Fuel, lb.	200	200
6	Reserve Fuel, lb.	5,410	5,430
7	Tankage Weight, lb.	3,865	3,890
8	Gross Weight, lb., 4 Engines Operating	103,371	103,716
9	Gross Weight at Takeoff, lb.	103,171	103,516
10	Gr. Wt., lb.; 1 of 4 Engines Can Be Shut Down	97,000	97,000
11	Four-Engine Fuel Weight, lb.	6,171	6,516
12	Average Gross Weight—Four Engines, lb.	100,085	100,258
13	Velocity/Fuel Flow Average	0.0212	0.0212
14	Nautical Miles for Four Engines	131	138
15	Gross Weight, lb., 3 of 4 Engines Open	97,000	97,000
16	Gr. Wt., lb.; 2 of 4 Engines Can Be Shut Down	52,000	52,000
17	Three-Engine Fuel Weight, lb.	45,000	45,000
18	Average Gross Weight—Three Engines, lb.	74,500	74,500
19	Velocity/Fuel Flow, Average	0.02805	0.02805
20	Nautical Miles for Three Engines	1,262	1,262
21	Gross Weight, lb., 2 of 4 Engines Operating	52,000	52,000
22	Final Gross Weight, lb.	49,271	49,316
23	Two-Engine Fuel Weight, lb.	2,729	2,684
24	Average Gross Weight, lb.	50,635	50,658
25	Velocity/Fuel Flow Ratio	0.0374	0.0374
26	Nautical Miles for Two Engines	102	100
27	Total Mileage, Nautical Miles	1,495	1,500

*7.5-percent of ΔFerry and Transport Total Fuels.

(C) The three-engine portion distance is the same for all iterations since the engine shutdown gross weights are always 97,000 pounds and 52,000 pounds. The average gross weight between these values is 74,500 with a resulting velocity/fuel flow ratio of 0.02805. Therefore, the fuel difference of 45,000 pounds times this value provides a distance of 1262 nautical miles.

(C) The fuel used for the final portion with two engines operating is obtained by subtracting the final arrival gross weight (empty weight with crew, reserve fuel, and tankage) from the 52,000-pound shutdown weight. The 49,271 pounds average gross weight yields a velocity/fuel flow ratio of 0.0374. The distance traveled, based on these values, was 102 nautical miles.

(C) The summation of the three portions is 1495 nautical miles. Based on this figure, a ratio is made of 1500/1495 times the fuel assumed of 53,900 pounds. This allows the new assumption of 54,200 pounds for the next iteration attempt.

(C) The use of the same procedure proved accurate within one-half mile with the following data being obtained:

	Four Engines	Three Engines	Two Engines
Average Gross Weight	100,258	74,500	50,658
Fuel Used	6,516	45,000	2,684
Velocity/Fuel Flow Ratio	0.0212	0.02805	0.0374
Total Nautical Miles	138	1,262	100
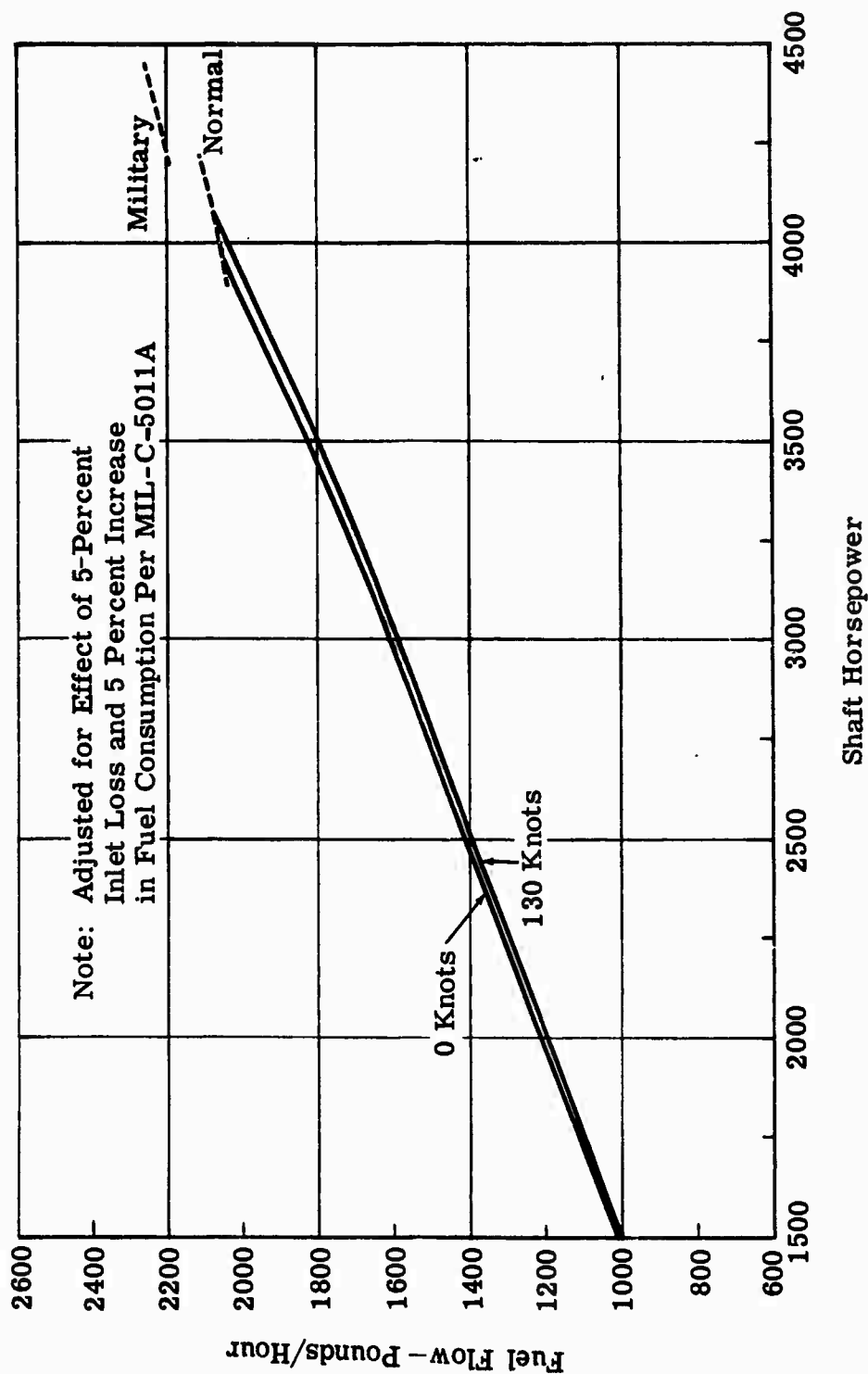
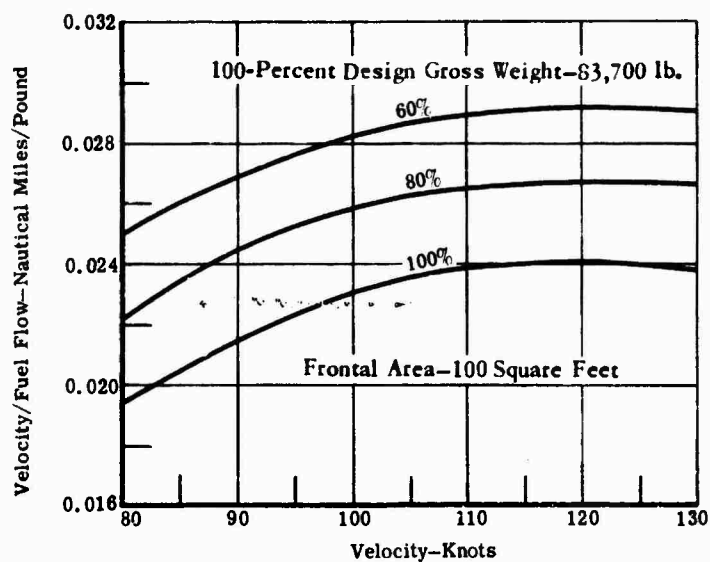Figure 116. (C) Adjusted Model 548-C2 Performance (U)

Figure 117. (C) Specific Range Versus Velocity and Gross Weight—Four Model 548-C2 Engines (U)

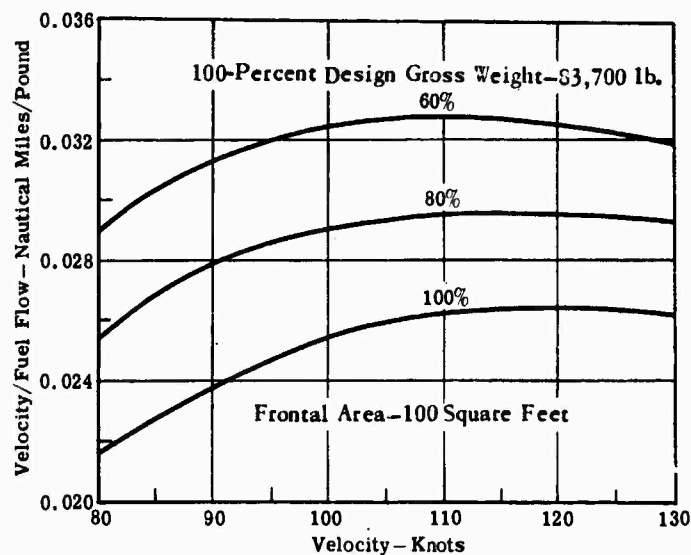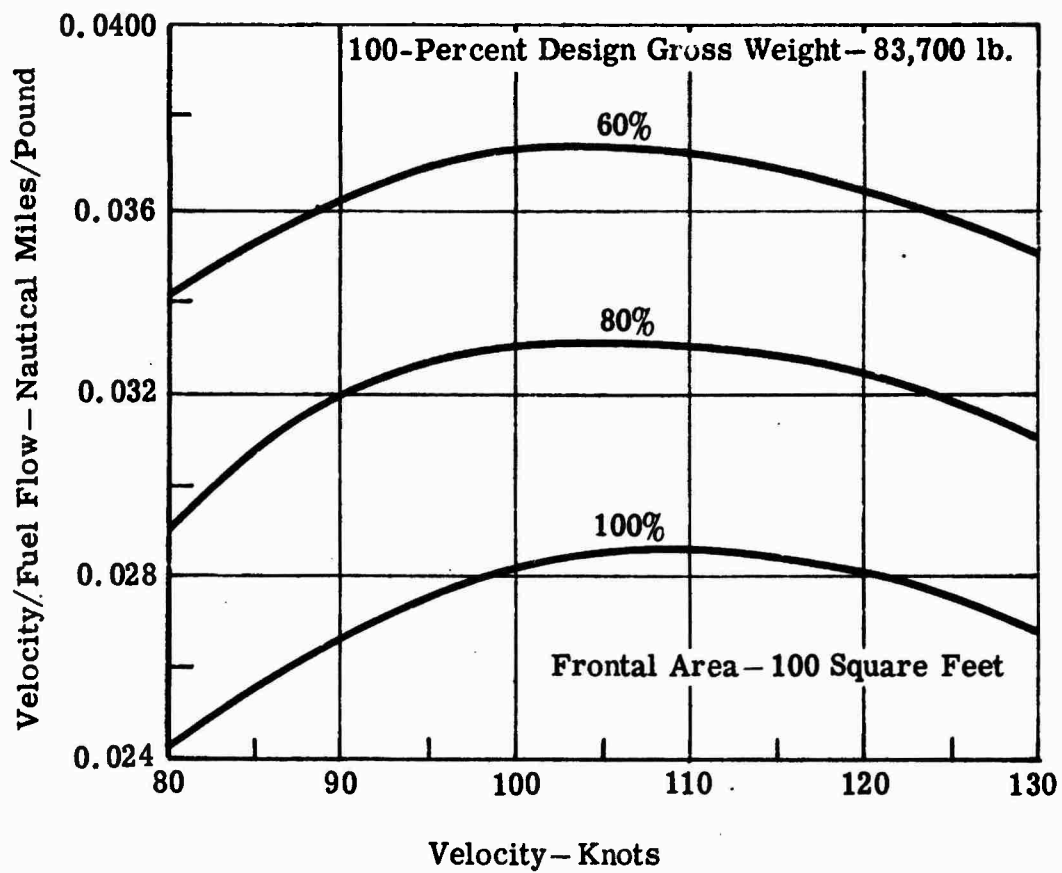**Figure 118. (C) Specific Range Versus Velocity and Gross Weight—Three Model 548-C2 Engines (U)**
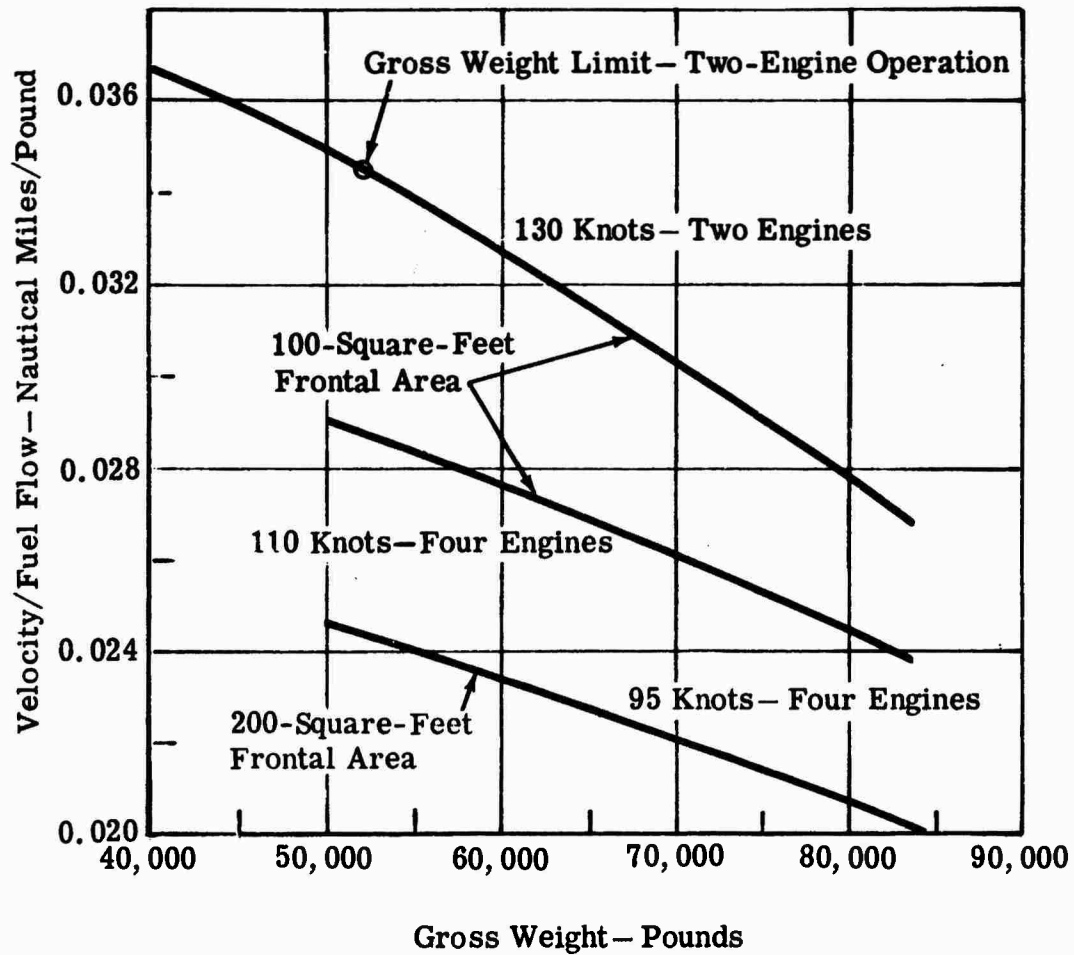
Figure 119. (C) Specific Range Versus Velocity and Gross Weight—Two Model 548-C2 Engines (U)

Figure 120. (C) Specific Range Versus Gross Weight for
Constant Velocity (U)
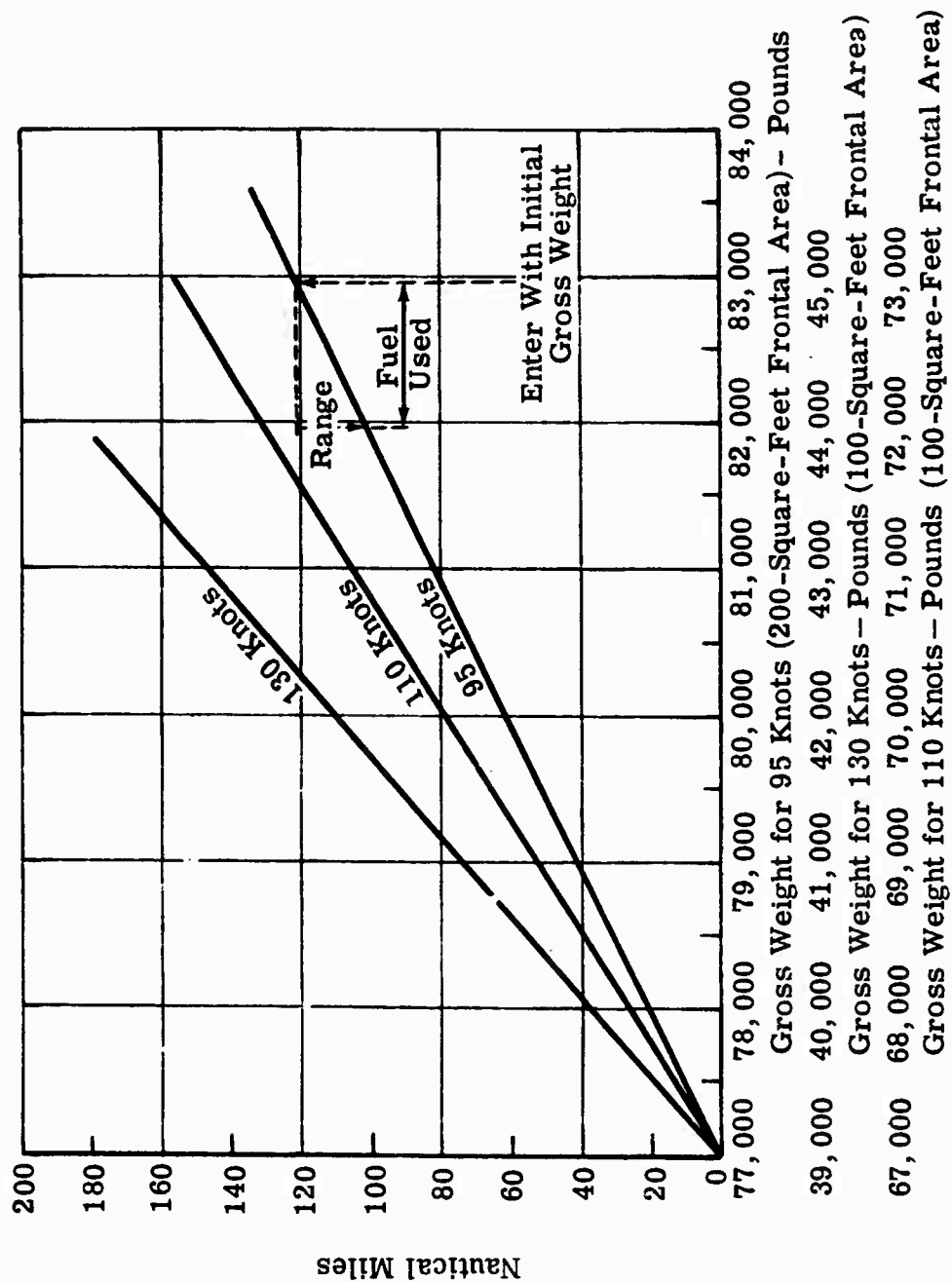
Figure 121. (C) Range and Gross Weight for Various Velocities (U)
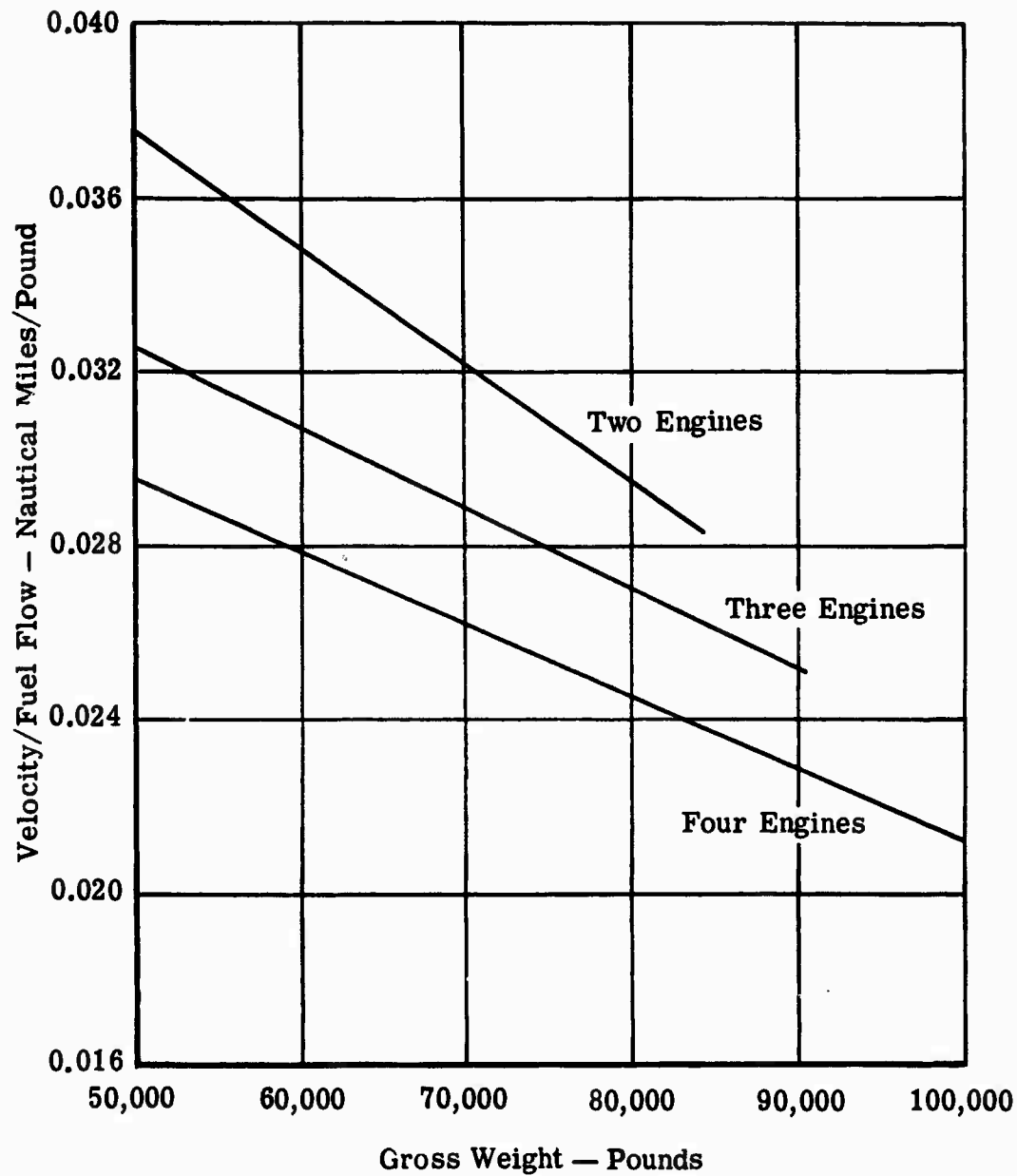
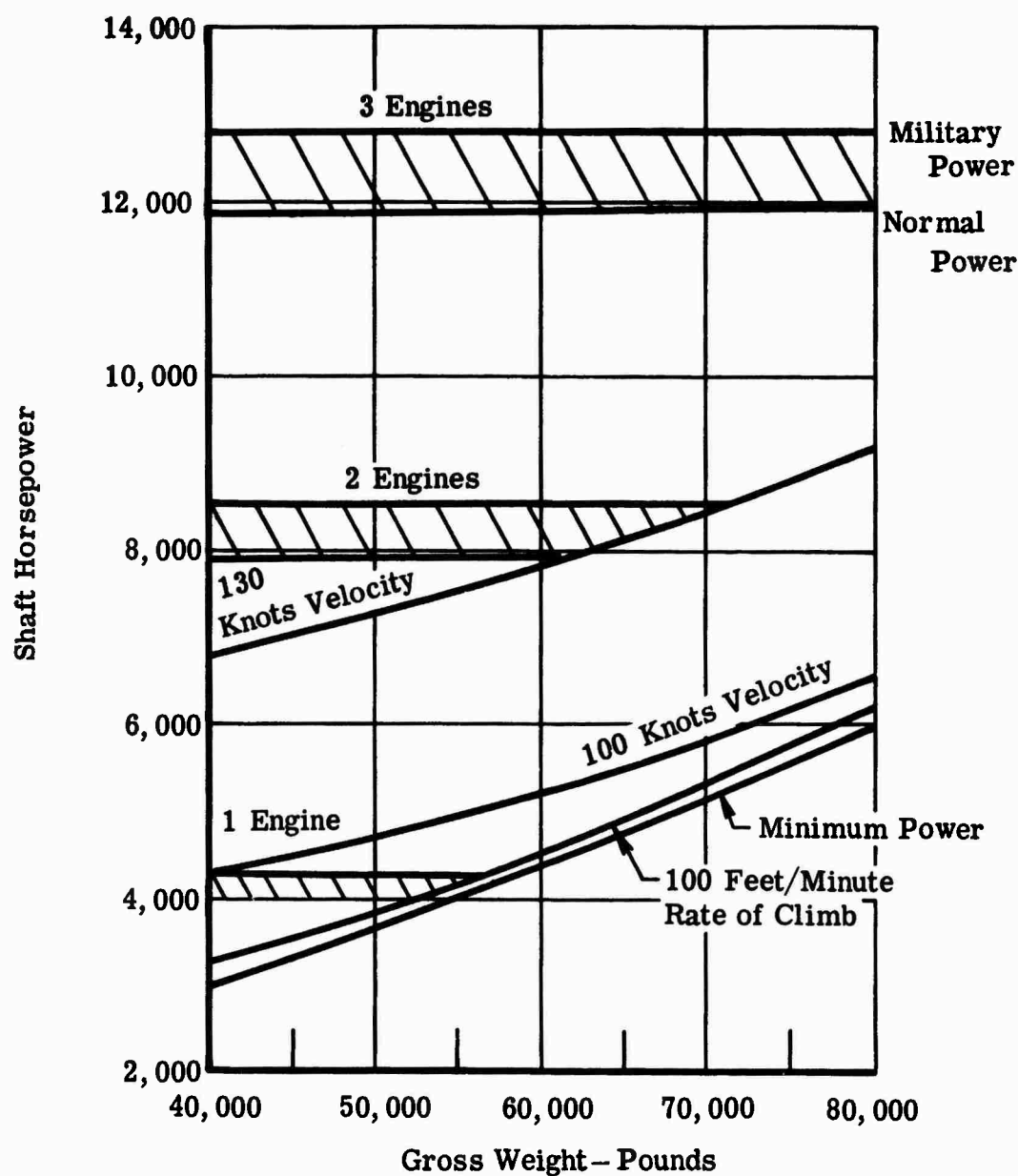Figure 122. (C) Specific Range Versus Gross Weight at Maximum Range –
Model 548-C2 Engines (U)

Figure 123. (C) Power Required and Available Versus Gross
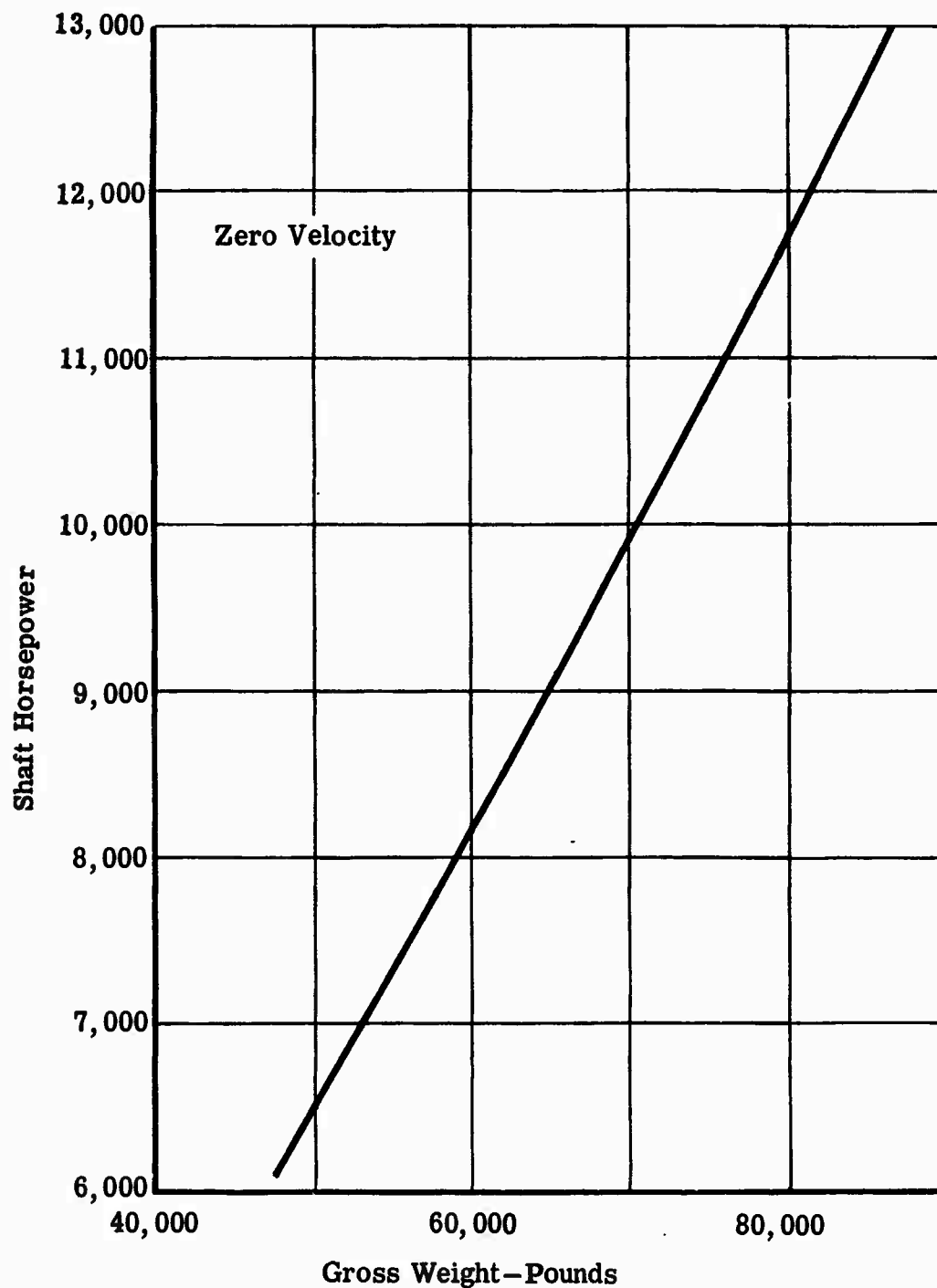Weight and Velocity (U)

Figure 124. (U) Hover (OGE) Shaft Horsepower Versus Gross Weight at Sea Level Standard Day Condition (Design Gross Weight = 83,700 pounds)

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